POWER

Vol. 44 NEW YORK, NOVEMBER 14, 1916 No. 20

The Engineer's Right to Recognition

Not many years ago the position of steam engineer was that of a hand workman; involving less skill and preparation than many of the manual trades, but commanding ordinary workman's wages.

The man in responsible charge of a modern power plant must possess qualifications that involve—in study, practice and intelligent application—as much time, expense and mental capacity as many of the professions. To an intimate knowledge of the complex system that he controls, which enables him to keep it going, and to jump at the right valve or switch or lever at the instant of trouble, he must add a knowledge of physics, electricity, refrigeration, chemistry and applied mechanics. He must be a good deal of an executive, keep informed of developments in a rapidly changing field, be on constant watch to foresee and head off interruption to the service, and produce power as cheaply as the other fellow—or show why not.

The change has come about so rapidly that the public and the employer have failed to grasp this evolution of the engine tender into the engineer. He continues to get a pitiful wage instead of an adequate salary.

The writer of the following letter is the representative of a numerous class. His is not an appeal for organized demand—the engineer likes to feel that he is a responsible part of his establishment—but a plea for recognition, social and pecuniary, for a class that suffers by being judged by what it was rather than by what it has become.

I have delayed sending in the renewal of my subscription to "Power" as I have been seriously thinking of quitting the work, and I will briefly tell you why.

I am rounding into my 45th year and the last 25 years—and you might say the best part of my life—have been spent in power-plant engineering; not working from 6 to 6 or just to do enough to hold my job but bringing all my energy and skill into the work. I have been a hard student all the time, reading the current magazines in the line, buying the best and latest books on steam and electrical engineering, studying fuel consumption and every phase of plant economy and applying what I learned when and wherever I could to the advantage of the company I was working for. At present I am taking the Street Railway Journal, Electrical World, Electrical Traction, Engineering Magazine and Power. My reference library on steam and electricity represents an investment of $400. in the best books I could buy and I use them too. All this outlay of time, money, energy and application for what? A chief engineer's job on duty 365 days out of the year subject to call 24 hours out of the day, and a salary of $150. per month.

Can you cite me to a trade or profession that requires the skill, knowledge, application and attention that our profession does and pays the beggarly wages that it does.

Our trade papers are continually urging their readers on to greater preparedness and fitness for their chosen work but rarely do you see an article dealing with the wage side of the question.

Love of the work and blind devotion to it does not alone suffice, and will not buy books, trade papers and keep the family above want.

Compression from a Mechanical Point of View

By F. R. Low

SYNOPSIS—The inertia and momentum effects of the reciprocating parts seriously modify the effective pressure upon the crankpin, especially with high rotative speeds. The article tells how the indicator diagram may be modified to show these effects and supplies tables to simplify the process.

Compression in a steam engine should be considered from two points of view, its effect upon the steam rate and its effect upon the running of the engine. Of the first of these phases much has been written; this article will deal with compression from the mechanical point of view.

Referring to the diagram, Fig. 2, when the piston, traveling in the direction of the arrow, arrives at the position indicated by the line *AB*, the pressure urging it forward has been reduced by expansion to that indicated by the height of the point *a*. At the same time there is on the other side of the piston, in the crank end of the cylinder, a pressure corresponding to the height of the point *b*, Fig. 3, the line *AB* being in the same relative position in both cases. If the diagrams are taken simultaneously, the solid back-pressure line of the crank end is being made at the same time as the solid forward pressure line of the head end, and the difference between these two solid lines (the forward-pressure line of the head-end diagram and the back-pressure line of the crank-end diagram) is a measure of the net, or unbalanced, steam pressure upon the piston.

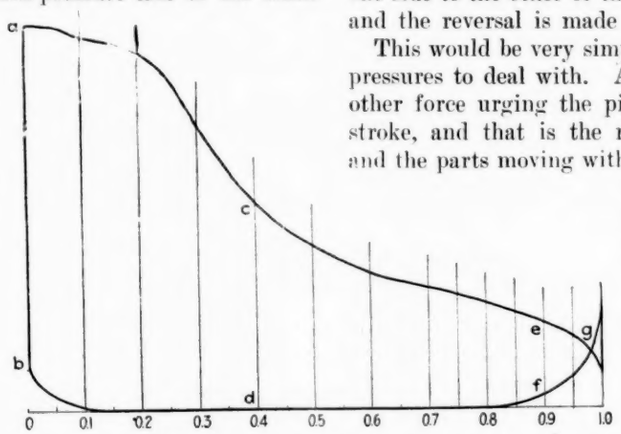
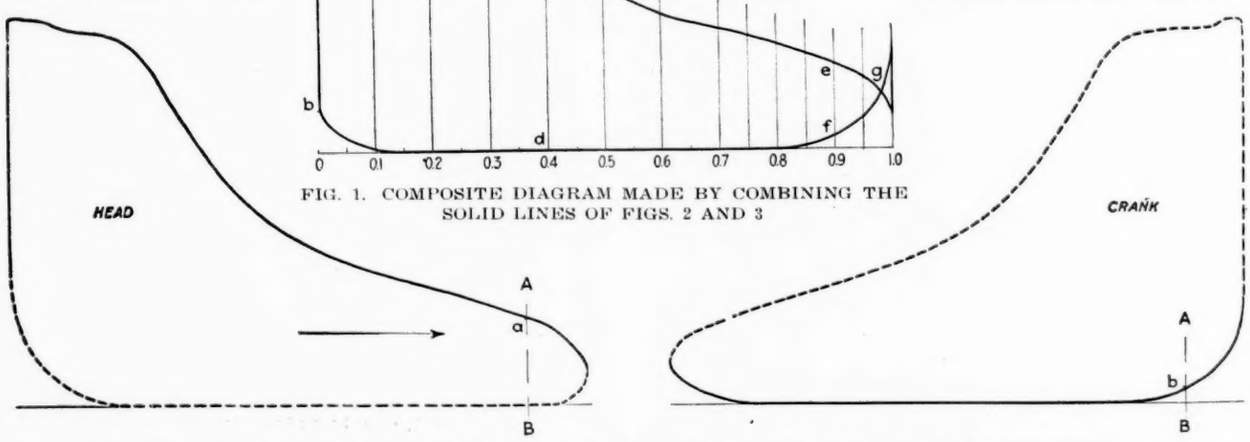


FIG. 1. COMPOSITE DIAGRAM MADE BY COMBINING THE SOLID LINES OF FIGS. 2 AND 3



FIGS. 2 AND 3. SIMULTANEOUS DIAGRAMS FROM THE HEAD AND CRANK ENDS OF CYLINDER RESPECTIVELY
The solid lines of both diagrams are being drawn at the sametime.

In a diagram like Fig. 1, composed of the two solid lines of the two diagrams in Figs. 2 and 3, the length of any vertical line, as *ab*, *cd*, *ef*, included between these lines is a measure of the net, or unbalanced, forward pressure in the cylinder at the position of the piston corresponding to the position of such vertical line. At the beginning of the stroke the unbalanced pressure will be *ab;* for the forward pressure was *a* and the back pressure at the same instant *b*. At the time the forward pressure was equivalent to *e* the back pressure was equivalent to *f* and the unbalanced pressure was proportional to the length of the line *ef*. At *g* the lines cross and the forward and backward pressures are equal. Up to this point the steam has been pushing on the crank—with a diminishing force it is true, but still pushing. The wristpin is pushed hard against the inside brass of the connecting-rod at *a*, Fig. 5; the inside brass of the other end of the rod is in close contact with the crankpin at *b*. The connecting-rod is in compression, and the shaft is pushed out against the front cheek of the main bearing at *c*. If this pressure were kept up until the end of the stroke and the steam valve opened when or before the crank reached the line of centers and admitted a considerable excess of pressure upon the other side, the sudden reversal of these forces would make a pound. By introducing compression, the forward force is balanced as at *g*, Fig. 1, and the excess pressure is gradually transferred to the other side. The pins and shaft float from one side to the other of their boxes under balanced forces, and the reversal is made without shock or jar.

This would be very simple if there were only the steam pressures to deal with. As a matter of fact there is another force urging the piston forward at the end of its stroke, and that is the momentum of the piston itself and the parts moving with it. Imagine the piston, piston rod, crosshead and connecting-rod of a good-sized engine wrapped about the crankpin in such a way that their centers of gravity lie in the center line of the pin, as in Fig. 4; then imagine the shaft to revolve at the speed at which the engine is supposed to run. The combined weight will exert a centrifugal force which varies as the square of the velocity and attains considerable magnitude as the rotative speeds

become high. This centrifugal force exerted by that mass of moving parts swung in a circle having a radius equal to the length of the crank, at the speed at which the engine runs, is the force which has to be exerted upon the piston to start the reciprocating parts into motion as the crank passes the line of centers. Even this force would start them off slowly—as slowly as the crank would move them as it made its first few degrees of departure from the center. Gradually the moving parts gain speed, and in order that they may do so, there must be continued application of force and absorption of energy; but although the speed increases until about the middle of the stroke, the rate of acceleration is greatest at first and decreases as the stroke progresses until at about midstroke, when the speed becomes greatest, the acceleration ceases. The moving parts then commence to slow down, and whereas they were a drag before and opposed by their inertia the pressure of the steam, keeping it from exerting its full force upon the crankpin, they now give up the energy that they have absorbed, their momentum urges them forward, and the crankpin is pushed ahead by the force which is required to stop them in addition to the pressure of the steam.

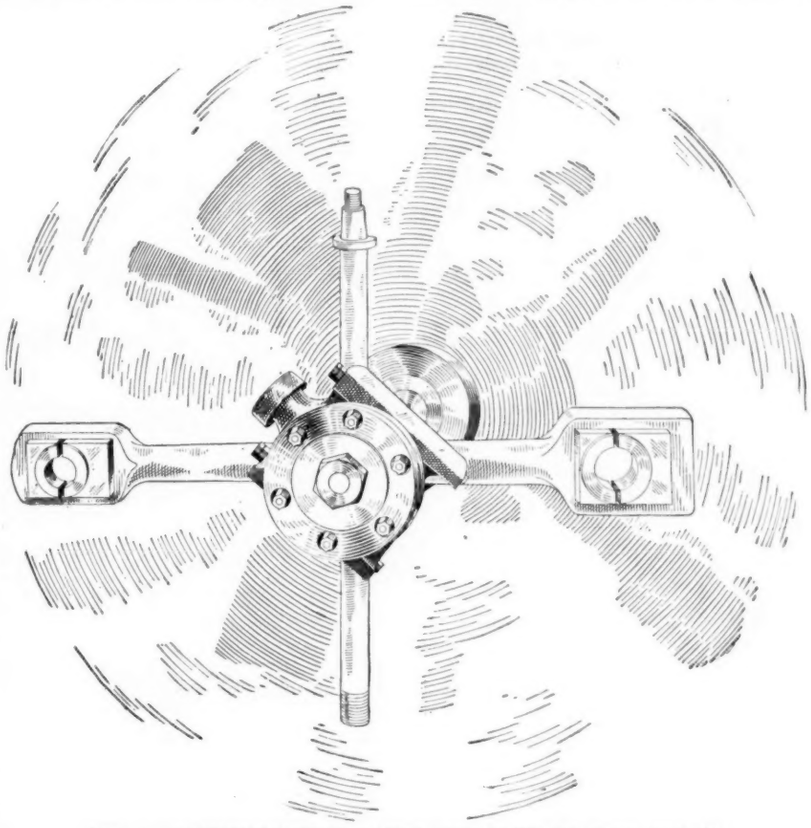


FIG. 4. RECIPROCATING PARTS WHIRLED BY THE CRANK

The centrifugal force exerted by the reciprocating parts whirled about on the crankpin is the force required to start those parts off the dead center at the same number of revolutions per minute.

If it were not for the disturbance introduced by the angularity of the connecting-rod, a diagram of this could be made very simply, as in Fig. 6, where *ab* represents the force required to start the reciprocating parts, the force required to further accelerate them diminishing along the line *ac*. Although the motion of the parts increases in velocity up to the center, the rate at which it increases diminishes until at the center (when the velocity is greatest) the rate of increase or the acceleration and the force required to produce it become zero, as indicated by the position of the point *c* upon the zero line. After the piston passes midstroke, it commences to slow down and their momentum urges the moving parts against the crank, exerting a pressure upon it in excess of that due to that of the steam. This pressure increases along the line *cd* and at the end of the stroke becomes proportional to *de*, which is equal to *ab*, the force required to get the parts into motion. The distance of any point upon the line *ad* from the line of centers is proportional to the pull or push of the moving parts on the crankpin in the direction of the piston's travel; *ab* is proportional to the force required to start the piston off the center, *ed* to the equal force required to stop it at the end of the stroke; the area *abc* to the energy absorbed in accelerating the velocity of the parts up to the maximum; and the equal area *ced* to the energy given up by the reciprocating parts in being brought to rest.

The connecting-rod with its angularity introduces some complication into this diagram as applied to an engine having a connecting-rod (as most engines do), but the complications have all been taken care of in the making of Table 1, by the use of which one may easily lay out a modification of Fig. 6 applicable to his own engine. This has been done in Fig. 7. Upon the line of zero pressures *OX* set off 20 equal divisions corresponding to the 20 equal portions of the stroke in the first column of Table 1. It has already been said that the force required to start the reciprocating parts at the dead-center

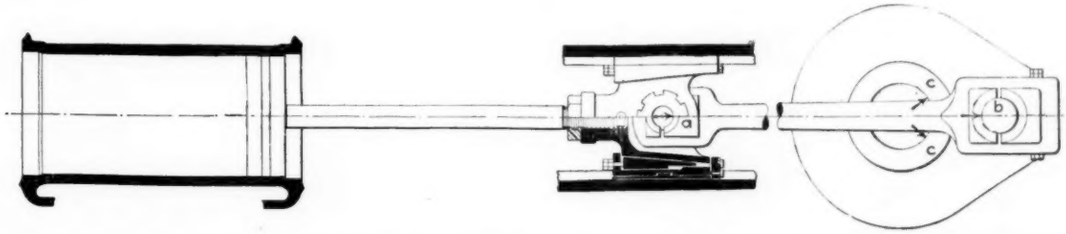


FIG. 5. SHOWING THE DIRECTION OF THE THRUSTS ON THE PINS AND SHAFT AT THE END OF THE STROKE

is equal to the centrifugal force that would be exerted by a weight equal to that of the reciprocating parts, swung in a circle having a radius equal to that of the crank and at the speed at which the engine runs. This total force divided by the area of the piston gives the pressure per square inch that must be exerted upon the piston to produce this force. This simmers down to the following[1]:

Rule: Multiply the weight in pounds, of the reciprocating parts, by 0.00001809, by the stroke in inches and by the square of the number of revolutions per minute, and divide by the square of the diameter of the cylinder in inches. The quotient will be the pressure per square inch that must be exerted upon the piston to start the reciprocating parts into motion at the given speed.

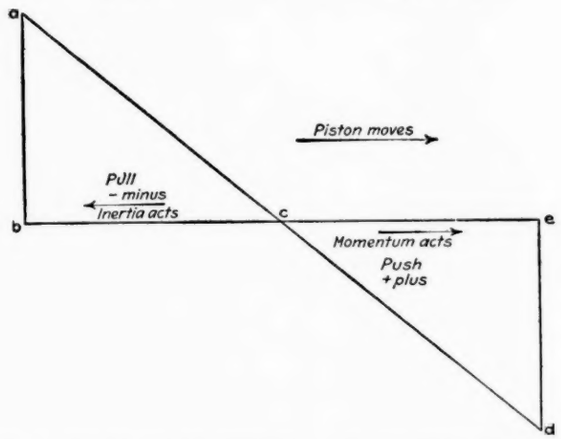


FIG. 6. DIAGRAM OF INERTIA AND MOMENTUM FORCE, UNCORRECTED FOR ANGULARITY OF CONNECTING-ROD

Example: The reciprocating parts of the high-pressure side of a Corliss cross-compound (the piston, piston rod, crosshead with its pin and connecting-rod) weigh 2,400 lb. The cylinder diameter is 24 in., the stroke 48

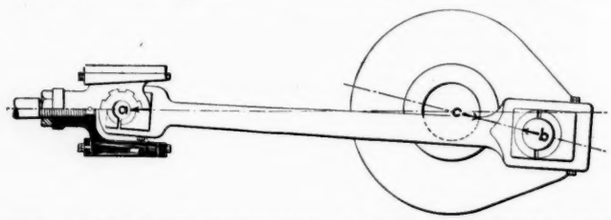


FIG. 8. SHOWING DIRECTION OF THRUSTS WHEN THE PIN PUSHES RECIPROCATING PARTS OFF CENTER

in. What will be the force required at the center to accelerate the moving parts when the rotative speed is 100 r.p.m.?

[1] F = Centrifugal force in pounds;
f = Centrifugal force per square inch of piston area;
w = Weight of reciprocating parts, pounds;
v = Velocity, feet per second;
r = Radius, feet;
s = Stroke, inches;
d = Diameter of cylinder, inches;
n = Revolutions per minute;
$\pi = 3.1416$;
g = 32.16.

$$v = \frac{s\pi n}{12 \times 60} \qquad v^2 = \frac{s^2\pi^2n^2}{720^2} \qquad r = \frac{s}{24}$$

substituting these in the formula for centrifugal force.

$$F = \frac{wv^2}{gr} = \frac{24ws^2\pi^2n^2}{720^2gs}$$

Divide this by the piston area $\frac{d^2\pi}{4}$ to get the centrifugal force per square inch of piston area

$$\frac{24ws^2\pi^2n^2}{720^2gs} \times \frac{4}{d^2\pi} = \frac{4 \times 24 \times 3.1416}{720 \times 720 \times 32.16} \times \frac{wsn^2}{d^2} = 0.00001809\,\frac{wsn^2}{d^2}$$

$$Force = \frac{\overset{\text{Weight}}{2{,}400} \times \overset{\text{Constant}}{0.00001809} \times \overset{\text{Stroke}}{48} \times \overset{\text{Sq. of r.p.m.}}{100 \times 100}}{\underset{\text{Sq. of cyl. diam.}}{24 \times 24}}$$

$$= 36.18\ lb.\ per\ sq.in.$$

To the same scale with which the indicator diagrams are taken set this value, 36.18, off on the first head-end ordinate (*am* of Fig. 7).

The force on the next ordinate (representing the position of the piston when the stroke is $^1/_{20}$, or 0.05, completed) is 0.914 of what it is on the center, as may be seen from the 10th column of Table 1, which gives the fraction of the initial pressure for each of the 19 intermediate positions of the piston of an engine having a connecting-rod 6 times the length of the crank. Multiplying the force at the commencement of the stroke by

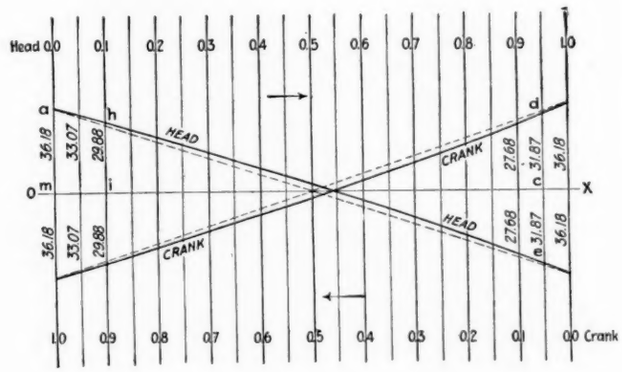


FIG. 7. DIAGRAM CORRECTED FOR ANGULARITY OF A ROD HAVING LENGTH OF 6 CRANKS

these values, the middle column of Table 2 is obtained. The plus and minus signs show whether the pressure is to be measured above or below the zero line, plus being below and minus above.

A little study of Table 1 will show that the value of the pressure for the crank diagram on any ordinate is

TABLE 1. RATIOS OF PRESSURES EXERTED BY THE RECIPROCATING PARTS IN THE LINE OF THE PISTON'S TRAVEL AT VARIOUS POINTS IN THE STROKE

Fraction of Stroke Completed	Connecting-Rod = 4 Cranks		Connecting-Rod = 4.5 Cranks		Connecting-Rod = 5 Cranks		Connecting-Rod = 5.5 Cranks		Connecting-Rod = 6 Cranks		Fraction of Stroke Completed
	Head	Crank	Head	Crank	Head	Crank	Head	Crank	Head	Crank	
0 00	—1 000	—1 000	—1 000	—1 000	—1 000	—1 000	—1 000	—1 000	—1 000	—1 000	0 00
0 05	—0 919	—0 869	—0 918	—0 874	—0.916	—0.877	—0 915	—0 877	—0 914	—0.881	0 05
0 10	—0 838	—0 744	—0 834	—0 751	—0 831	—0.757	—0.829	—0 762	—0.826	—0.765	0 10
0 15	—0 754	—0 623	—0 744	—0.633	—0 745	—0.641	—0 741	—0 647	—0.738	—0.652	0 15
0 20	—0 670	—0 506	—0 663	—0 518	—0 657	—0 527	—0 652	—0 535	—0 648	—0 541	0 20
0 25	—0 583	—0 393	—0 575	—0 406	—0 568	—0 416	—0 563	—0 424	—0 558	—0 432	0 25
0 30	—0 495	—0 283	—0 486	—0 298	—0 478	—0 309	—0 471	—0 318	—0 466	—0 325	0 30
0 35	—0 406	—0 177	—0 395	—0 192	—0 386	—0 203	—0 378	—0 213	—0 372	—0 220	0 35
0 40	—0 314	—0 074	—0 302	—0 088	—0 292	—0 100	—0 284	—0 109	—0 277	—0 117	0 40
0 45	—0 221	+0 027	—0 208	+0 012	—0 197	+0 001	—0 188	—0 008	—0 181	—0 016	0 45
0 50	—0 125	+0 125	—0 111	+0 111	—0 100	+0 100	—0 091	+0 091	—0 083	+0 083	0 50
0 55	—0 027	+0 221	—0 112	+0 208	—0 001	+0.197	+0 008	+0 188	+0 016	+0 181	0 55
0 60	+0 074	+0 314	+0 088	+0 302	+0 100	+0 292	+0 109	+0 284	+0 117	+0 277	0 60
0 65	+0 177	+0 406	+0 192	+0 395	+0 203	+0 386	+0 213	+0 378	+0 220	+0.372	0 65
0 70	+0 283	+0 495	+0 298	+0 486	+0 309	+0 478	+0 318	+0 471	+0 325	+0.466	0 70
0 75	+0 393	+0 583	+0 406	+0 575	+0 416	+0 568	+0 424	+0 563	+0 432	+0 558	0 75
0 80	+0 506	+0 670	+0 518	+0 663	+0 527	+0 657	+0 535	+0.652	+0.541	+0 648	0 80
0 85	+0 623	+0 754	+0 633	+0 744	+0 641	+0 745	+0 647	+0 741	+0 652	+0 738	0 85
0 90	+0 744	+0 838	+0 751	+0 834	+0 757	+0 831	+0 762	+0 829	+0 763	+0 826	0 90
0 95	+0 869	+0 919	+0 874	+0.918	+0.877	+0 916	+0.877	+0 915	+0 881	+0 914	0 95
1 00	+1 000	+1 000	+1 000	+1 000	+1 000	+1 000	+1 000	+1 000	+1 000	+1 000	1 00

the same as it is on the same ordinate for the head-end diagram, only with a different sign. The ordinate 0.1 of the head-end scale is the ordinate 0.9 of the crank-end scale, and it will be seen that the tabular value in both cases is 0.826, only that the 0.826 of the pressure on the center is — in the case of the head end and + in the case of the crank end. All that needs to be done therefore to make the crank-end diagram after that for the head end has been drawn is to set off on all the ordinates distances (as *cd*) on one side of the base line equal to the

distance (as *ce*) of the point through which the head-end diagram passes on the other side of the zero line.

In Fig. 9 these pressures have been transferred to the composite diagram, Fig. 3. The unbalanced steam pressure on the head center, or ordinate 0, is proportional to *ab*, Figs. 1 and 9. The force required to accelerate the moving parts is proportional to *am*, Figs. 7 and 9, There is available for exerting pressure upon the crankpin only the difference, *mb*, Fig. 9. Setting off on the corresponding ordinates the distances, as *am*, *hi*, of the points at which the line of the head-end diagram crosses the ordinate from the center, below the forward pressure line when the sign is minus, above it when the sign is plus, and connecting the points so placed, a diagram of the pressure upon the crankpin as modified by the momentum and inertia of the reciprocating parts is obtained. The effective pressure for a little more than the first half of the stroke has been reduced by the pressure required to accelerate the reciprocating parts. This becomes zero at *e*, and for the rest of the stroke the pressure is increased by that required to bring the parts again to rest, leaving the terminal pressure at *c* in excess of the pressure *d* attained at the end of the stroke by compression. The area *ame* represents the energy absorbed in getting the parts into motion and the equal area *ecf* that given up by them to the crankpin in being brought to rest.

In the case of the engine making the diagram shown in Fig. 9, the reciprocating parts are in the position shown by Fig. 5 when the crankpin arrives at the center. If there were a prompt opening of the steam valve, the direction of stress would be suddenly reversed, all the lost motion would be taken up with a slam and the engine would pound. If the admission of steam is delayed, however, until the crank gets around, as in Fig. 8, so as to push the crosshead away, keeping the connecting-rod still in compression the gradual increase of effective pressure which follows a late admission as modified by the absorption of some of this pressure in overcoming the inertia of the moving parts may allow the change in direction of the excess force and the change in the condition of the connecting-rod from compression to extension to be made gradually enough to avert a pound; but it would be better to increase the compression as shown by the dotted line on Fig. 9 until it crosses the line of effective forward pressures, as at *g*, before the piston reaches the center and allow the steam valve to open with the customary amount of lead.

If this engine had been run at 75 revolutions instead of 100, the diagram (produced in the same way from the values in column 2 of Table 2) would be like Fig. 10. At this slower speed the effect of the momentum and inertia is less and the terminal effective pressure *c* is less than that, *d*, attained by compression. The com-

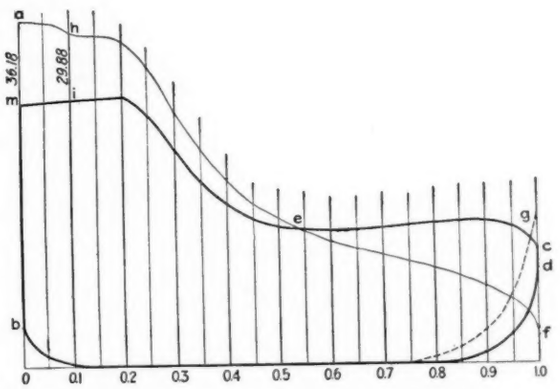


FIG. 9. DIAGRAM FIG. 1 CORRECTED FOR INERTIA AND MOMENTUM EFFECTS AT 100 R.P.M.

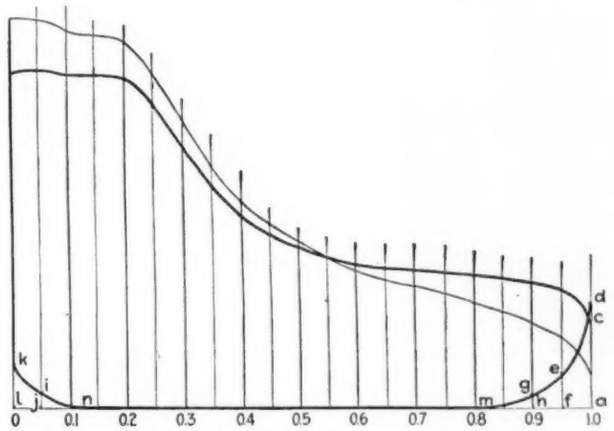


FIG. 10. DIAGRAM FIG. 1 CORRECTED FOR INERTIA AND MOMENTUM EFFECTS AT 75 R.P.M.

In both cases the diagrams with the lighter lines are the originals, those in heavier lines being the corrected forms

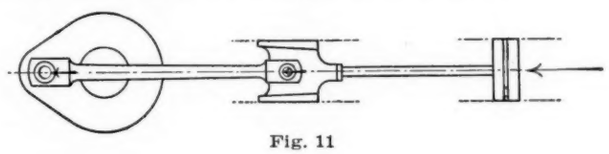


Fig. 11

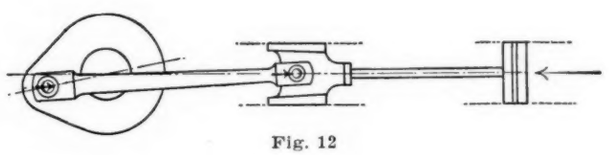


Fig. 12

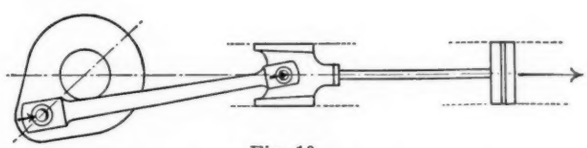


Fig. 13

FIGS. 11 TO 13. SHOWING DIRECTION AND REVERSAL OF THRUST ON CONNECTING-ROD AND PINS

TABLE 2. MAGNITUDE AND DISTRIBUTION OF INERTIA EFFECTS IN A SLOW-SPEED AND A HIGH-SPEED ENGINE

Fraction of Stroke Completed, Head End	Corliss Engine Diam. of cyl. 24 in. Stroke 48 in. Wt of recip parts 2400 lb. Speed 75 r.p.m.	Corliss Engine Diam. of cyl. 24 in. Stroke 48 in. Wt. of recip parts 2400 lb. Speed 100 r.p.m.	Lentz Engine Diam. of cyl. 20 in. Stroke 24 in. Wt of recip. parts 2454 lb. Speed 200 r.p.m.	Fraction of Stroke Completed, Crank End
0.00	—20.35	—36.18	—106.54	1.00
0.05	—18.60	—33.07	—97.38	0.95
0.10	—16.81	—29.88	—88.00	0.90
0.15	—15.02	—26.70	—78.63	0.85
0.20	—13.19	—23.44	—69.04	0.80
0.25	—11.36	—20.19	—59.45	0.75
0.30	—9.08	—16.86	—49.65	0.70
0.35	—7.57	—13.46	—39.63	0.65
0.40	—5.64	—10.02	—29.51	0.60
0.45	—3.68	—6.55	—19.28	0.55
0.50	—1.69	—3.00	—8.84	0.50
0.55	+0.33	+0.58	+1.70	0.45
0.60	+2.38	+4.23	+12.47	0.40
0.65	+3.48	+7.96	+23.44	0.35
0.70	+6.61	+11.76	+34.63	0.30
0.75	+8.79	+15.63	+46.03	0.25
0.80	+11.01	+19.57	+57.64	0.20
0.85	+13.27	+23.59	+69.47	0.15
0.90	+15.57	+27.68	+81.51	0.10
0.95	+17.93	+31.87	+93.86	0.05
1.00	+20.35	+36.18	+106.54	0.00

pression of Fig. 3 is therefore quite enough for this engine running at 75 revolutions, but at 100 it should be increased so as to balance the forward effective pressure before the end of the stroke is reached.

There are still a couple of factors to be considered. One of these, the effect of different areas on either side of the piston (as by a rod on the crank side and none on the head), can be readily calculated. Subtract the area of the piston rod from that of the piston and divide by the area of the piston. The result will be the fraction of their apparent values to which the pressures of the indicator diagram from the crank end of the cylinder should be reduced on account of the lesser area upon which they act. For example, the 24-in. cylinder

Fig. 14 is a reproduction of a pair of diagrams from a 20x24-in. engine running at 200 r.p.m., which gives it the same piston speed as the Corliss of the foregoing examples when running at 100. The diagrams in lighter line are the composites, modified for the effects of momentum and inertia. Starting with the crank end, the initial pressure, or the pressure on the crank center, is represented by the height of the point *a*, but it requires a force proportional to the distance *ab* to start the reciprocating parts off the center. The distance of *c* above the zero line represents the back pressure, hence *ac* the unbalanced forward pressure, which is insufficient, by the difference *bc*, to overcome the inertia of the reciprocating parts. The piston has therefore to be moved off the cen-

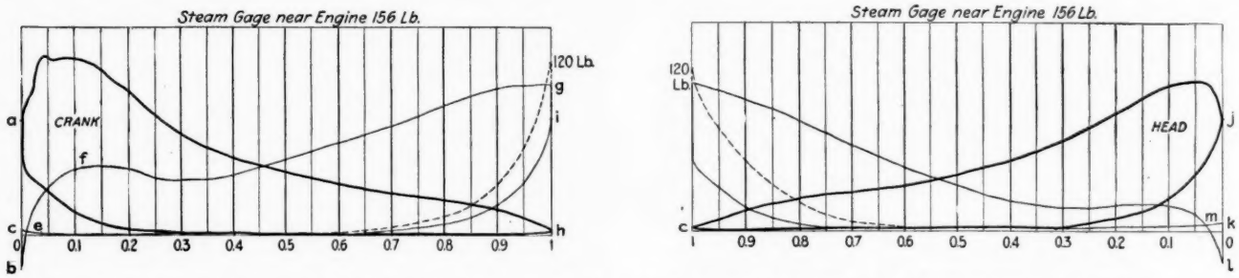


FIG. 14. DIAGRAMS FROM LENTZ ENGINE, THOSE IN HEAVY LINES AS THEY CAME FROM THE INDICATOR, THOSE IN LIGHT LINES SHOWING INERTIA AND MOMENTUM EFFECTS

of the example given had a 4¼-in. rod. The area of a 24-in. circle is 452.39 sq.in., that of a 4¼-in. circle 14.186:

$$\frac{452.39 - 14.186}{452.39} = 0.969$$

To correct for the reduction of area of the crank side of the piston on Fig. 10 therefore, all that would be necessary would be to make each distance, as *ad, ef, gh, ij, kl,* from the line of zero pressures to the back-pressure line (which came from the crank-end diagram) 0.969 of what it measured on the diagram itself. This would make the compression line *dm* and the release line *kn* slightly lower.

The effect of the other factor, friction, is impossible to calculate, but it is not of great magnitude—equivalent to about one pound average pressure per square inch of piston area. Lowering the forward-pressure line or raising the back-pressure line by this amount will correct the diagram for it in the ordinary case.

At these slow speeds the effect of momentum and inertia is comparatively small, but it increases as the square of the number of revolutions per minute. The effect on an engine running at 200 revolutions per minute will be four times as great as that on an engine running at half this speed, other conditions being the same.

ter by the crankpin. When the crank reaches the crank-end dead-center (the end of the head-end stroke), there is 2.5 lb. of residual pressure (*c*, Fig. 14) in the head end and a pressure of 106.5 lb. per sq.in. due to momentum. The total force acting on the crankpin is therefore 106.5 + 2.5 = 109 lb. per sq.in. on a 20-in. piston, or 314.16 × 109 = 34,243.44 lb., or over 17 tons exerted in the direction to put the rod into compression, as shown in Fig. 11, with nothing to counteract it but the compression in the crank end.

The crankpin, carried around by the energy stored in the flywheel, pushes the piston off the center, and in doing so keeps its pressure upon the same side of the box, as in Fig. 12, keeping the rod still in compression, notwithstanding its change of direction. At *e*, Fig. 14, the effective forward pressure equals the counterpressure. The forward and backward force upon the pin are balanced. Up to this point the rod has been in compression as the crank, through it, pushed the piston off the center. At this point the force exerted through the rod becomes zero, and gradually, along the line *ef*, the effective pressure becomes greater in the direction of the piston's movement and the piston commences, after passing the point *e*, to pull the crankpin, putting the connecting-rod into extension. This condition is shown at Fig. 13.

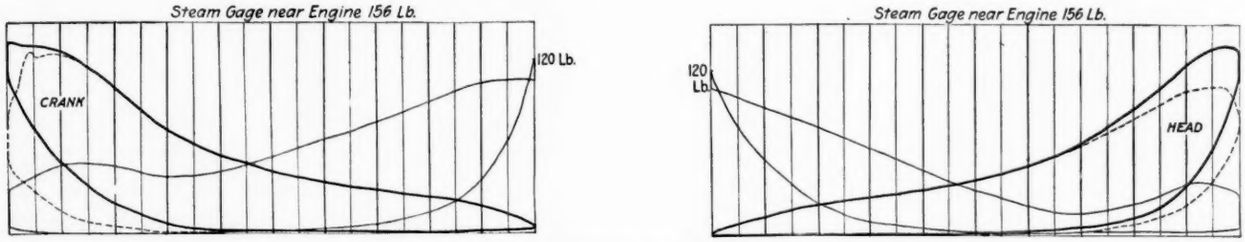


FIG. 15. DIAGRAM WHICH THE SAME ENGINE OUGHT TO MAKE WITH MORE COMPRESSION AND LEAD

At the head center there is a force due to the combined action of the residual pressure and momentum proportional to hg; but there is a compression pressure proportional to gi, the difference being equal to about 25 lb. per sq.in., or $314.16 \times 25 = 7{,}854$ lb., or less than 4 tons.

If pressure were now suddenly admitted into the head end there would be a pound, as all the lost motion is exposed in the position to be taken up; but the pressure in the head end of the cylinder is j, in the crank end k, the unbalanced pressure in the head end jk and the pressure necessary to start the piston off jl.

There is not pressure enough to be felt through the anvil offered by the reciprocating parts; the pin pulls the piston off, keeping the rod in extension until the pressures become balanced at m, and the direction of force is gradually shifted as described in connection with the crank-end diagram. Compression to 120 lb., as indicated by the dotted compression lines in Fig. 14, would make the diagrams from the cylinder look like those in Fig. 15. The filling of the clearance by compression and the earlier opening of the steam valve would probably result in the realization of more initial pressure (the lines of the present diagrams, Fig. 14, are shown dotted on Fig. 15), and the unbalanced-force diagram indicated by the lighter lines has all the essentials for quiet operation. There is less unbalanced pressure on the bearings, a simple easy reversal of stress while the piston runs into a cushion, and the diagram area gained by realization of greater initial pressure and prompt admission ought to more than make up for that lost by the greater compression, so that the capacity of the engine will not be reduced.

Readers will find the study of their own engines in the light of such diagrams interesting and profitable. To recapitulate the process:

1. Divide the diagram into 20 equal parts.
2. Number the dividing lines as in Fig. 9, progressing in the direction in which the piston travels.
3. Compute by the rule given the force per square inch of piston area necessary to start the engine off the center.
4. Multiply the value so found by the factors given in Table 1 for the ratio of connecting-rod to crank used on your engine.
5. To the same scale as that with which the diagrams were taken lay off the values as found on their respective ordinates, measuring downward from the forward pressure line when the sign is minus and upward when the sign is plus.
6. Connect the points thus located to form the upper line of the diagram and trace in the back-pressure line of the indicator diagram from the opposite end of the cylinder for the lower line.

Slot Wedges and Band Wires—General sentiment appears to consider slot wedges preferable to band wires from an insulation standpoint, yet there is a feeling that they are likely to break and come out. There should be no such trouble if the thickness of the wedge is ample and the grooves holding the wedges are deep. An interesting test of a slot-wedge construction was recently observed on a 1,600-r.p.m. machine. The shunt field was accidentally opened where no overload protection was provided. The consequent overspeeding burst the rear armature head band, bent the coil heads and wrecked the field coil. Since the motor had been run on the test bench at 2,700 r.p.m., with no loosening of the head band, it seemed reasonable to assume that the runaway speed considerably exceeded this value. Careful examination of the slot wedges showed that none was broken or split. The use of such wedges requires a deeper slot for the same winding, hence a somewhat larger motor.

The Durabla Water-Gage Safety Appliance

For the safety of the fireman and to increase the visibility of the water level in the gage-glass, W. G. Werbeck, 216 Fisher Building, Chicago, designed the appliance shown in the accompanying illustrations. Mr.

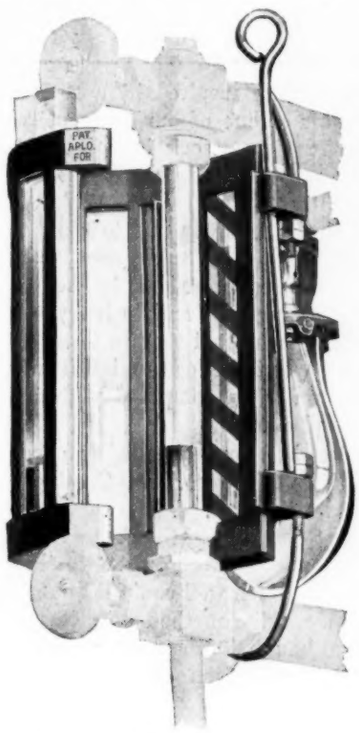


FIG. 1. CASE SWUNG OPEN

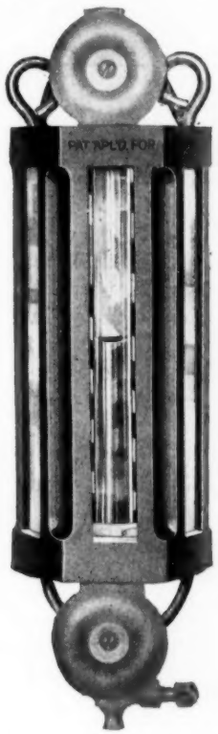


FIG. 2. CASE CLOSED

Werbeck is the Western sales manager for the Durabla Manufacturing Co., of New York City. The device is made for marine and stationary boilers and may be adapted to different sizes and types of water gage. It consists of an aluminum envelope with four flat glass windows 3/8 in. thick, front, back and sides, the back portion of the casing being attached to the pipes leading to the gage-glass by staples top and bottom. The front part of the casing, containing three windows, is hinged on either side to the rear section, the hinge pins being easily removable, so that the case may be swung open in either direction.

On the rear window of the envelope are painted red diagonal stripes, which in the water-filled portion of the glass appear to slope in the opposite direction. This color scheme, in connection with a black frame and an incandescent lamp back of the device, makes the water level unusually distinct and visible from a considerable height or a long distance horizontally. The safety feature is also important, for if a gage-glass should break, the inclosing envelope protects the fireman from hot water, steam or pieces of broken glass. An appliance of different design but for the same purposes is made for locomotive boilers.

Improving Insulation—A good baking varnish is a cheap method of improving insulation. An extra dip and bake can be secured at an outlay of 0.36 per cent. of the total cost of the motor. At 550 volts and below, mechanical strength and length of life are the most important features of good insulation.

Excess Air in Fuel Combustion

By Morgan B. Smith

SYNOPSIS—A brief exposition showing the true meaning of fuel losses due to the use of excess air in the combustion of coal in terms of dollars and cents.

The following paragraphs were inspired by a remark to the writer by the operator of one of our large power plants to the effect that "in view of our immense production the losses in our power plant do not cut any serious figure in our operation." A few minutes' talk with this man convinced him that it was "up to him" to look into this matter of losses, and subsequent work showed him that the largest losses were due to the use of too much air in the boiler furnaces and to leaky settings. This same man finally admitted that in terms of dollars and cents to his employer these losses might easily mount into large figures even when compared with the immense production of the plant.

In this plant many devices had been installed to check the plant operation, including CO_2 recorders, Orsat gas analyzers and draft gages over the fires. These were kept in operation in a perfunctory sort of way, the charts being carefully filed away and as carefully forgotten once they found their way into the files. Such well-equipped plants are not uncommon, nor indeed the method of utilizing such forms of apparatus, although in them we have the means of detecting errors in operation and the means for the correction of bad conditions.

Losses Due to Excess Air

Probably the greatest cause of poor combustion lies in the theft of thousands of heat units from the boiler surfaces by too great volumes of air admitted to the combustion chambers. Fig. 1 shows theoretically perfect combustion of a unit weight of coal, just enough air being used to convert the coal into available heat units without loss. Such a condition in practice is not possible. In this case assume that each unit of air finds its way to a particular unit of coal, giving perfect contact of air with coal. In practice this is not possible, the fact being that we are obliged to use approximately 12 per cent. of excess air to get somewhere near perfect mixing of gas with air. This then constitutes the starting point for comparing good with bad conditions of combustion.

Fig. 2 shows the best practical conditions for the combustion of coal. Conditions of load and plant equipment may show that a slightly increased amount of excess air gives the best results. In this case we are supplying to the furnaces 12 per cent. of air in excess of that theoretically required by the combustible matter in the coal, showing 15 per cent. CO_2 in the resulting flue gases and a minimum amount of oxygen and nitrogen. In this case we may assume the preventable fuel loss to be zero although the total loss is about 12 per cent. of the fuel burned in the furnaces.

Fig. 3 shows fairly good conditions involving a total loss of 18 per cent. of the total fuel used, with 6 per cent. preventable. Fig. 4 shows poor conditions, the total loss amounting to about 26 per cent. of which 14 per cent. is preventable. Fig. 5 shows very poor conditions, the total loss being about 49.5 per cent. of which fully 37.5 per cent. is preventable.

Strange as it may seem Figs. 4 and 5 show conditions which are common in spite of the general awakening among plant operators to the opportunities for betterment of their conditions.

In Table 1 the net cost of coal is considered to be the initial cost plus the loss per ton and from this basic figure we have calculated the losses in dollars and cents and have attempted to show the purchasing power represented by this loss. This is figured in two ways; namely, in terms of the money loss per year of 300 working days per a daily fuel requirement of 50 tons of coal, and in terms of this money capitalized at 6 per cent. per annum, the latter representing the purchasing power of the money lost if taken to the bank for an operating loan subject to 6 per cent. interest. This is the *true loss* to the plant owner.

TABLE 1. DOLLARS AND CENTS LOSSES DUE TO POOR COMBUSTION

	Conditions Shown in Fig. 1	Fig. 2	Fig. 3	Fig. 4	Fig. 5
Initial cost of coal per ton, dollars..	2.40	2.40	2.40	2.40	2.40
Preventable loss per ton, cents	....	....	14.4	33.6	90.0
Net cost of coal per ton. Initial cost plus loss, dollars	....	....	2.544	2.736	3.30
Preventable yearly loss. Basis: 50 tons coal burned per day for 300 days, dollars.....	....	....	2,160	5,040	13,500
Preventable loss if capitalized at 6 per cent., dollars.	....	....	36,000	84,000	225,000
Preventable loss will purchase the labor of one man at $4 per day for	....	....	540 days (1.5 yr.)	1,260 days (3.5 yr.)	3,375 days (9.25 yr.)
Same capitalized at 6 per cent. is.....	....	....	24.6 yr.	57.5 yr.	154.1 yr.
Preventable yearly loss on 50 tons coal daily at net cost will buy coal to operate the plant for	....	....	17 days	37 days	81.8 days
Same if capitalized at 6 per cent. is..	....	....	283 days (0.8 yr.)	616 days (1.7 yr.)	136.3 days (3.74 yr.)

In the case of conditions shown in Fig. 4, where flue gases show 7 per cent. CO_2, a very common condition in small plants, this capitalized loss amounts to fully $84,000 per year, an amount sufficient to purchase considerable new apparatus for the plant; to pay a good salary to a man capable of improving conditions or to purchase fuel for the plant for a very considerable period of operation.

The conditions of Fig. 5 are extreme but nevertheless are found in many small plants, the losses running into thousands of dollars per annum and the purchasing power so great as to be staggering to the plant owner when he comes to a full realization of their meaning.

How then may one know how much excess air is used? The first step is to analyze a fair, average sample of the fuel it is expected to burn. From this one can calculate the theoretical amount of air necessary per pound of coal, adding thereto about 12 per cent., giving the minimum weight of air required.

The next step is to equip the plant with the apparatus which will show continuously the amount of air actually used. For this purpose there are the Orsat type of gas

FIGS. 1 TO 5. SHOWING THE EFFECT OF EXCESS AIR ON COMBUSTION. PERCENTAGES REFER TO VOLUME.

Fig. 1—Theoretical combustion, no loss of fuel. Fig. 2—Excellent practice, 12 per cent. loss, 0 per cent. preventable. Fig. 3—Fairly good practice, 18 per cent. loss, 6 per cent. preventable. Fig. 4—Poor practice, 26 per cent. loss, 14 per cent. preventable. Fig. 5—Very poor practice, 49.5 per cent. loss, 37.5 per cent. preventable.

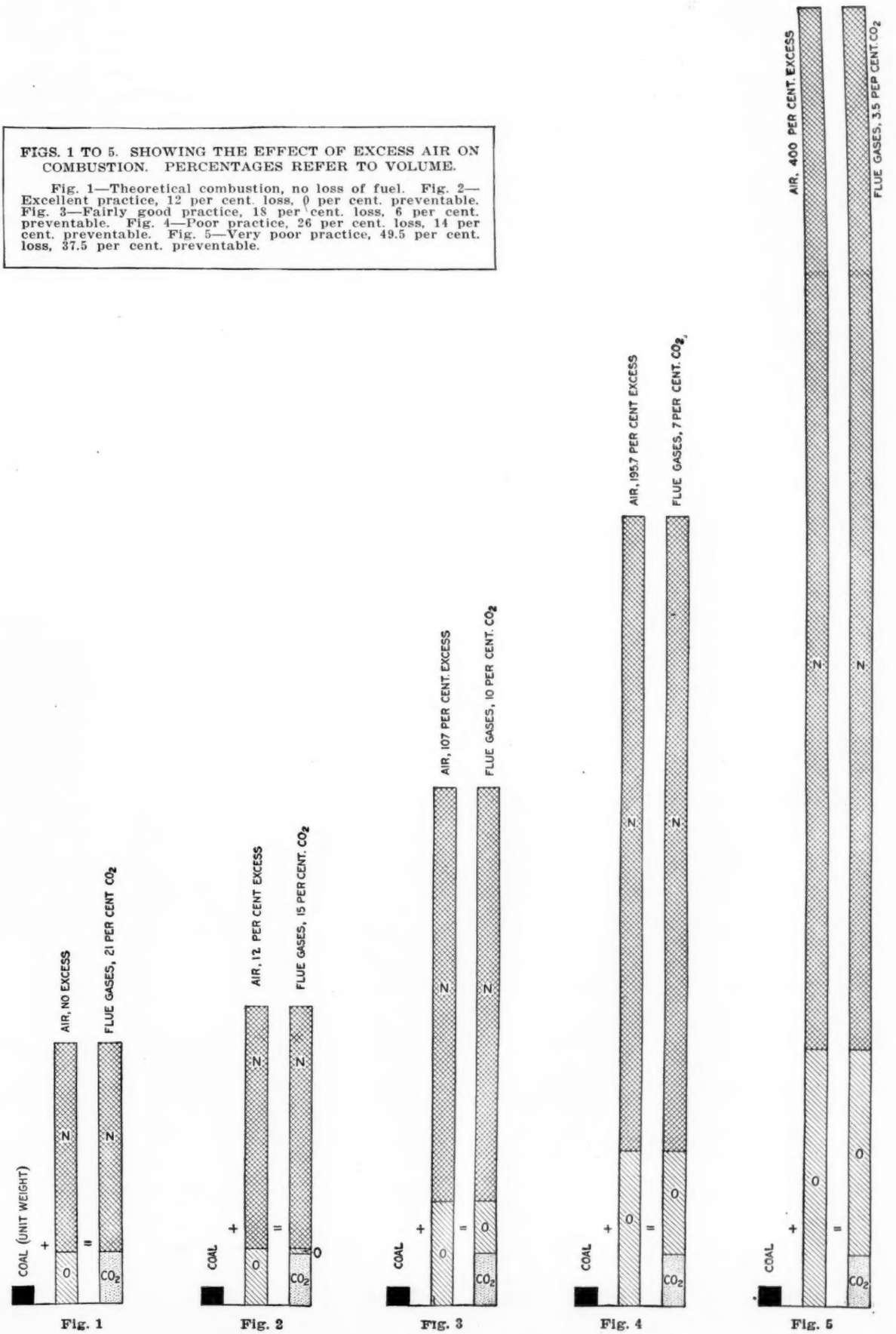

COAL (UNIT WEIGHT)
AIR, NO EXCESS
FLUE GASES, 21 PER CENT CO_2
COAL
AIR, 12 PER CENT EXCESS
FLUE GASES, 15 PER CENT. CO_2
COAL
AIR, 107 PER CENT. EXCESS
FLUE GASES, 10 PER CENT. CO_2
COAL
AIR, 195.7 PER CENT EXCESS
FLUE GASES, 7 PER CENT. CO_2
COAL
AIR, 400 PER CENT. EXCESS
FLUE GASES, 3.5 PER CENT. CO_2

Fig. 1 Fig. 2 Fig. 3 Fig. 4 Fig. 5

analyzers, the CO_2 recorder, the differential draft gage, and, lastly and indirectly, the electric pyrometer.

The gas analyzers show the result of existing conditions and do not give a hint of the causes unless oxygen is determined in addition to the CO_2 in the flue gases.

Draft gages, suitably installed at the rear damper, over and under the fuel bed, show existing conditions and give the causes of good or bad results.

Pyrometers, installed over the fires or at the top of the first pass in water-tube boilers, afford a measure of the output of the boiler which must be taken into account.

It is well known that high CO_2 can be attained at very low boiler outputs, a fact which firemen are not slow to learn, especially when bonuses are paid for high

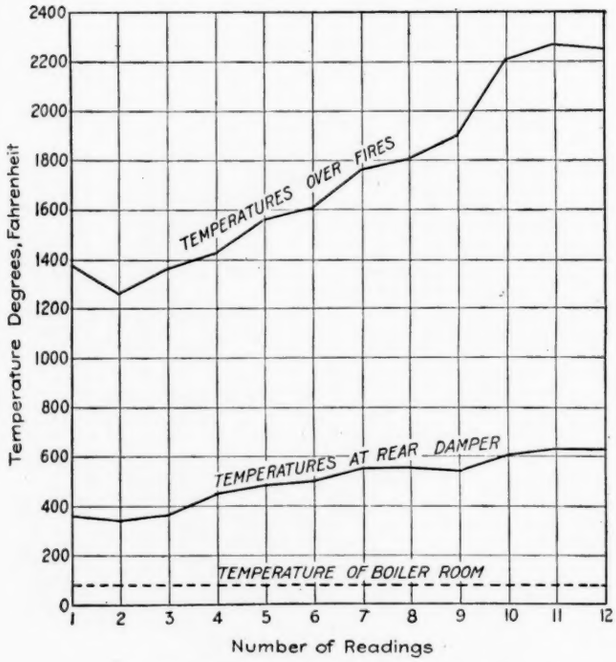


FIG. 6. RELATION OF GAS TEMPERATURES

content of CO_2 in the flue gases. It is equally practical to attain high outputs with fairly high CO_2.

Why is it not wise to adopt in power-plant operation the slogan of modern medical practice that "Prevention of disease is better than it's cure"?

The writer has often made the remark that, given a set of draft gages and a pair of pyrometers, he would not much care whether CO_2 could be determined or not. He does not mean to disparage the use of the CO_2 recorder or of the Orsat analyzer in the least. They have their value and can be used to advantage when combined with proper draft and temperature regulation.

In this case the use of steam-flow meters or boiler-feed meters is recommended, since they afford direct measure of the boiler output.

With pyrometers installed as stated it is possible to maintain a uniform temperature drop through the boiler, this drop being a measure of the amount of heat absorbed by the boiler and therefore of the evaporation, provided all heating surfaces, such as tubes and drums, are kept clean inside and out and so can absorb the maximum amount of heat units.

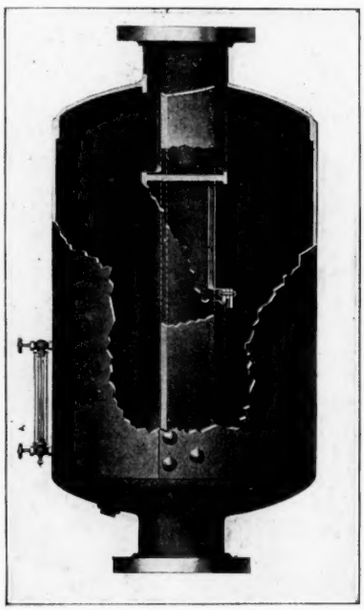
In Fig. 6 are shown the results of proper regulation of temperatures compared with previously existing conditions. This is the case where fires were not kept hot and where the stack gases were needlessly low in temperature. By running the fires much hotter the gases to the stack were raised about 200 deg. in temperature yet the boiler output was greatly increased. Before the change in operation the boiler surfaces were absorbing little heat; after the change they did so much more work that the increased loss to the stack was more than offset by the increased steam production. The installation of pyrometers made this a possibility and afforded means to maintain the latter conditions.

Temperatures are the result of draft conditions and quality of fuel; likewise the CO_2 content of the flue gases. Draft therefore, is of most vital interest in the economical operation of plants and one will do well to keep close watch upon this factor of combustion.

Welderon Receiver Separator

A new design of receiver separator is shown herewith, in which the longitudinal seam of the shell is welded and in which the heads are welded to the shell and to the continuous pipe passing through it. It is made by the Elliott Co., 6908 Susquehanna St., Pittsburgh, Penn.

The separator consists essentially of a part of the main steam header that is provided with a diaphragm and with inlet and outlet ports and surrounded with a steel-plate body. Referring to the illustration, the pipe is shown divided into two portions by a diaphragm that forces the steam and entrained moisture from the steam header into the separator body through the inlet port, the area of which is only slightly in excess of the pipe area, thus causing the steam and water to enter the receiver at a high velocity. As the separator has a large holding capacity, the steam comes almost to rest upon entering the receiver chamber, but the water, owing to its density, is carried to the bottom, out of the path of the steam. The area of the outlet being in excess of the pipe area eliminates the entering friction of steam with its corresponding drop in pressure.



SEMI-SECTIONAL VIEW OF VERTICAL SEPARATOR

The steam circulates around the receiver and passes into the steam header through the outlet port shown at the left. The long baffle plates extending at an angle from each side of the continuous pipe prevent moisture from being carried from the inlet to the outlet port or from creeping along the pipe.

Oil Containing Paraffin will emulsify, in warm water, more readily than if without, and on being allowed to stand in a glass vessel will show a whitish film between the oil and water when cooled.

Developing More Water Power on Genesee at Rochester

A hydro-electric development, novel because made under unusual topographical conditions and also because it is made within the residential limits of a large city, is now under way by the Rochester Railway and Light Co. at Rochester, N. Y. The Genesee River in its flow through the city has four falls totaling 227 ft. in head, in addition to 22 ft. of intervening rapids. For some years the company has had two power developments at the two lower falls, known as station No. 5 at the lower falls and station No. 15 at the middle falls about 1,400 ft. upstream. The former station, shown in the foreground in Fig. 1, develops 8,800 hp. from the 94-ft. head by taking the water from behind a low diversion dam rounding the crest of the falls. The middle falls station, seen in the far background of Fig. 1 (indicated in plan, Fig. 2), develops 2,700 hp. from the 27-ft. head by taking water from behind a gravity section concrete dam 6 ft. high. The new project consists in supplanting the upper dam by a new movable dam 16 ft. high which will elevate the pond 13 ft., and in building a tunnel carrying the water from this pond clear down to the face of the lower cliff, thus providing a low water head of 137 ft. in the new project, a gain of 13 ft., due to the extra head of the new upper pond and to the utilization of 3 ft. of rapids now wasted between the two falls. This increase of head and the increased efficiency of the new turbines increase the available power to about 32,000 hp. In addition the increase of height of the dam makes possible so large a storage behind it that for handling short peak loads the output may be raised to 48,000 hp. The new dam is made up of movable sections, as shown in Fig. 3, because the river is subject to flashy floods. Its maximum recorded flow is 45,000 cu.ft. per sec.; its minimum is 150 cu.ft. per sec., so that auxiliary steam power is required. When the new project is completed the old station No. 15 will be abandoned and the entire power concentrated in new station No. 5, old station No. 5 being removed. The data on the new development can best be appreciated by the following tabulation: Turbine Installation — Estimated cost, $1,056,000 for present installation of 32,000 hp. Ultimate capacity of plant, 48,000 hp. Turbines are I. P. Morris Co. vertical Francis type, developing 16,000 hp. at a head of 130 ft. Speed, 180 r.p.m. Weight of one turbine, 272,000 lb. Governors are I. P. Morris Co. double floating-lever governors, water operated. Weight of one governor, 43,000 lb. Generators are Westinghouse 12,500 - k v. - a., three-phase, 60-cycle, 11,500-volt. A combined efficiency of turbine and generator of over 88% is expected. Mounted on each generator is a 250-volt 100-kw. direct - connected exciter. It requires 57,500 cu.ft. of air per min. to cool each generator at full load. The rotating parts of the exciter, generator and turbine and the hydraulic thrust of the turbine are carried by a Kingsbury bearing capable of supporting 100 tons.

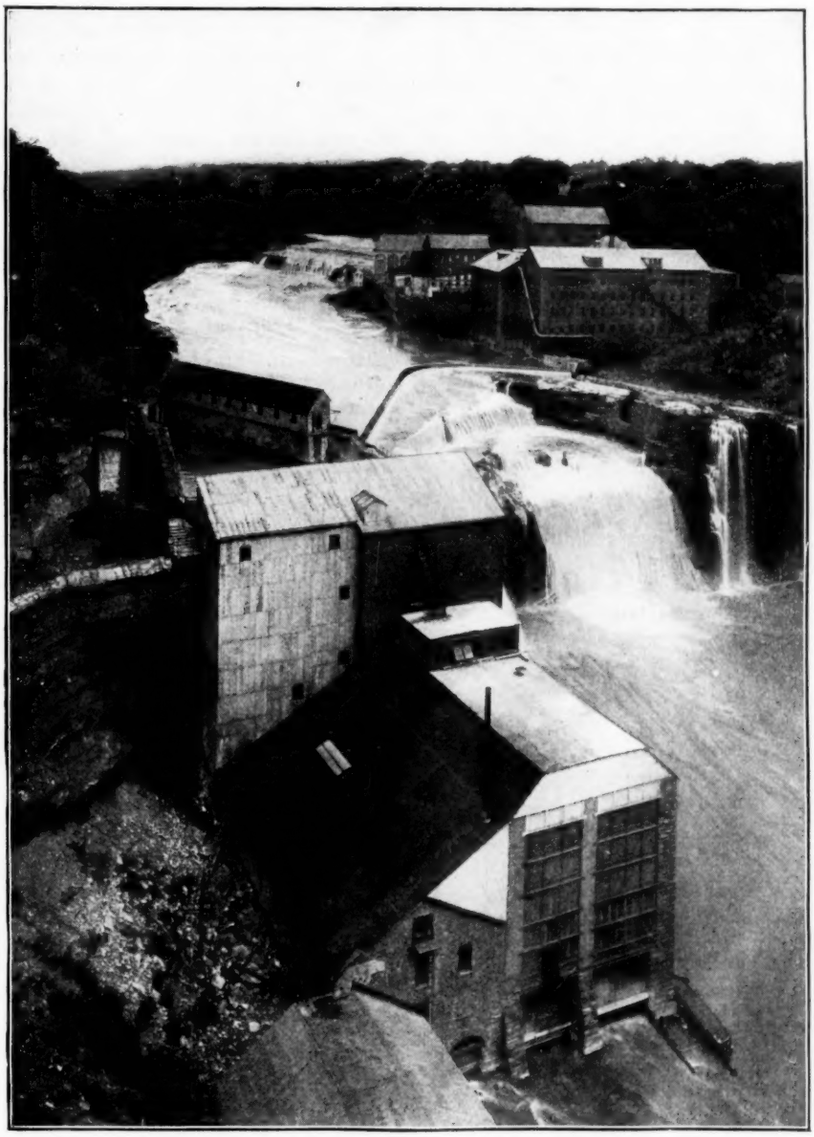


FIG. 1. LOOKING UP THE GENESEE RIVER FROM DRIVING PARK BRIDGE, ROCHESTER, N. Y.

Old Station No. 5 in foreground; Station No. 15 and site of new dam in far background

Power House—Reinforced-concrete and steel building. Present length, 90 ft. Will be increased to 125 ft. when

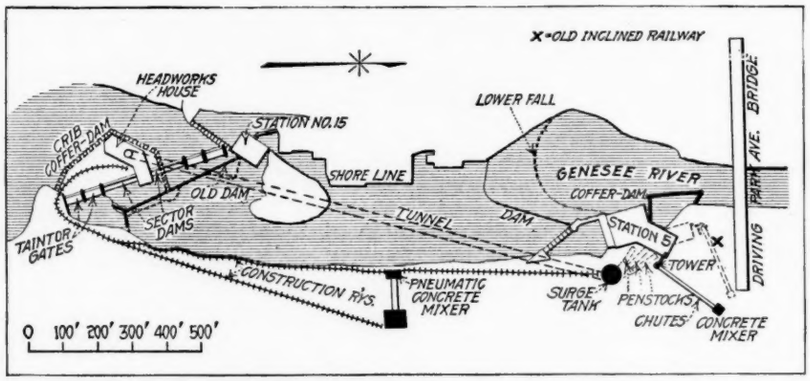


FIG. 2. PLAN OF NEW POWER DEVELOPMENT OF ROCHESTER RAILWAY AND LIGHT CO. ON GENESEE RIVER

third unit is installed. Width, 75 ft. Height, 112 ft. above rock. Will serve as a generating and substation and replace old Stations Nos. 5 and 15.

Valves—Flow of water is controlled by Johnson valves contained in the penstock. These valves serve as venturi meters for measuring the flow of water.

Penstocks—Three penstocks are carried out of the rock. One is closed off with a steel bulkhead. These are 13 ft. in diameter and built of ¾-in. plate. They extend about 30 ft. into solid rock. Tunnel—Horseshoe shape. Area, 317 sq.ft. Capacity, 3,900 cu.ft. per sec., with a velocity of 12.3 ft. per sec. Dam—Movable steel structure with concrete base and piers. Total length of spillway, 400 ft., made up of four Taintor gates each 50 ft. long and 16 ft. high and two sector gates each 100 ft. long and 16 ft. high. Each Taintor gate weighs about 40 tons and each gate hinge weighs 3 tons, two hinges to each gate. Each sector gate weighs 150 tons. Each sector gate hinge weighs 90 tons, being continuous and 100 ft. long. Intake—Located in center of river. Has 16 openings 10 ft. wide by 12 ft. high, eight on each side of building; controlled by steel gates operated by traveling electric hoist from above. Rubbish kept out by outer curtain wall which extends 4 ft. below low water and by steel trash racks inside buildings. Pond—Total storage from El. 375.5 to El. 390 equals 23,000,000 cu.ft. of water, which will produce 25,000 kw.-hr. of energy.

Surge Tank—A steel-plate differential surge tank 60 ft. in diameter, 48 ft. deep, with riser 18 ft. in diameter, built under R. D. Johnson's patent.

Construction Work To Be Done

The construction elements of the new development consist in the erection of a new movable dam at the middle falls, the driving of a shaft in the river at that point and a tunnel 1,400 ft. long to the face of the cliff at the lower falls, the sinking of a surge tank shaft just above the lower falls but in the bank to the east of the river and the building of the power plant alongside of old station No. 5 at the lower falls.—*Engineering News.*

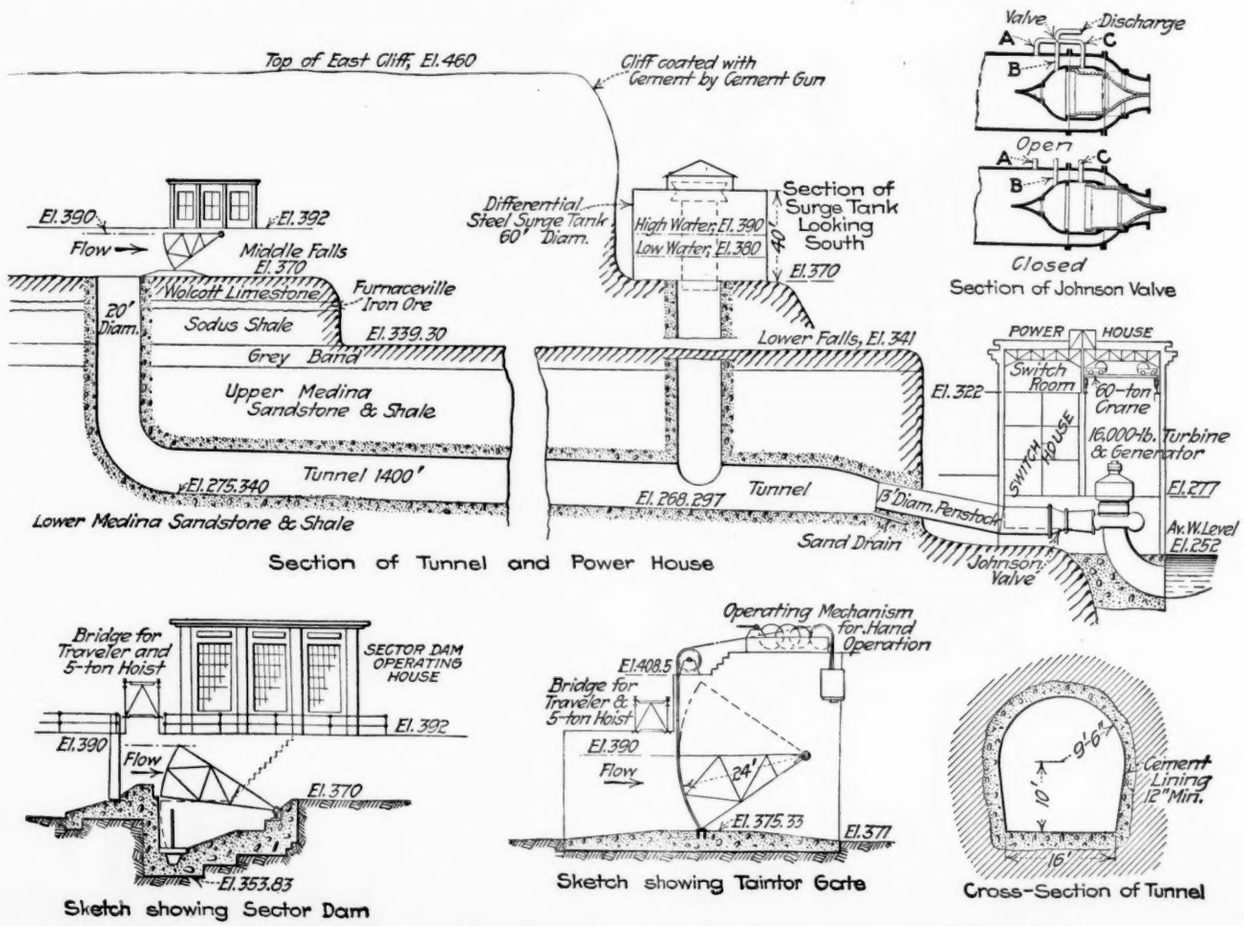


FIG. 3. DETAILS OF NEW HYDRO-ELECTRIC DEVELOPMENT AT ROCHESTER, N. Y.

Synchronous Converters--III

By F. A. Annett and Gordon Fox

SYNOPSIS—This is the final article of the series treating on rotary converters and takes up the different methods of voltage regulation used on synchronous converters and the methods of starting these machines.

The theoretical voltage ratios of rotary converters are slightly modified by several design and operative factors. The wave shape of the impressed voltage has an influence. A peaked wave has a higher peak value than a flat-topped wave, the effective values being the same. The direct voltage, corresponding to the peak value, is thus higher with a peaked impressed voltage. The ratio of pole arc to pole pitch of the converter influences the shape of the generated counter voltage and thus affects the voltage ratio. Armature resistance plays a role by introducing voltage drop, most pronounced, of course, under load. Brush position has an influence, just as it affects the voltage delivered by a direct-current generator.

The amount of field excitation influences the voltage ratio, although its more important action is in a modification of impressed voltage, as later explained. With lagging current the counter voltage is less than the impressed alternating pressure and the direct voltage is less than normal. With leading current the counter voltage exceeds the normal impressed alternating voltage and the direct voltage is greater than normal.

It was shown in Part I (Oct. 17 issue) that the ordinary single-phase, two-phase or diametrical six-phase converter has a voltage ratio of about 0.707, the alternating voltage being about 0.707 of the direct voltage. A three-phase or six-phase double-delta converter has a voltage ratio of about 0.612.

Voltage Control of Rotary Converters

It is usually desirable that the direct voltage delivered by a rotary converter be either variable or adjustable. For railway and similar industrial service an approximate and variable voltage is desired, corresponding to the rising characteristic of an overcompounded direct-current generator. For lighting service an accurately adjustable and fairly steady voltage is usually sought. There are a number of ways in which the voltage delivered by a converter may be controlled. The more prominent methods of today involve regulation of the alternating voltage impressed at the converter terminals. This regulation may be secured in several ways, namely: By feeder-voltage regulators inserted in the alternating-current leads; by a booster alternator connected in series with the alternating-current side of the converter; by use of reactance in connection with wattless current drawn by the converter.

The feeder-voltage regulator is a special type of transformer comprising essentially a primary winding connected across the line and a secondary winding in series with the line. The primary winding is wound on a rotatable drum, while the secondary winding is stationary. Voltage control is secured by turning the movable element through an angle, thus changing the phase relation of the secondary series voltage to line voltage, which it is vectorially added to or subtracted from. The feeder-regulator method of voltage control has been widely applied, but is being superseded by the other mentioned methods. The regulators involve extra equipment which, if automatically controlled, is quite complicated. As the series coils of the regulator carry the load current, they must be capable of standing overloads and short-circuit currents.

The booster converter comprises a standard shunt-wound converter with an auxiliary alternator connected upon the same shaft. The auxiliary unit is connected in series on the alternating end between the converter and the transformers. Adjustment of the voltage of the auxiliary machine varies the alternating impressed voltage and thus varies the direct voltage to a like degree. The auxiliary machine must have the same number of poles as the converter itself. It may be either of the rotating-field or rotating-armature variety. Units of the revolving-field type are usually mounted outside of the bearing pedestal, while those of the revolving-armature type are usually mounted inside the bearing pedestal, quite close to the converted armature, as in Fig. 24. The voltage of the booster may be either additive or subtractive, depending upon the polarity of the booster's poles, which is controlled by the reversible rheostat, thus affording considerable range without excessive size. The auxiliary unit must carry the full load current since it is connected in series. The booster voltage may be automatically controlled and may be changed from boost to buck and vice versa without interruption. This type of converter now finds extensive use for almost every class of service, especially in large-sized machines.

The point was brought out in Part II (Oct. 31 issue) that it is possible to adjust the power factor of a rotary converter by changing its excitation so that it will draw from the line a lagging or leading magnetizing or demagnetizing wattless current. A lagging current, in passing through a reactive circuit, causes a voltage drop. A unity power-factor current does likewise but to a lesser degree. A leading current passing through a reactive circuit causes the voltage at the receiver end to be higher than at the feeder end.

Voltage Control by Reactance

In a rotary-converter installation the transformer windings are used for introducing reactance into the circuit. Converter excitation is used as a means to draw wattless current through this reactance. Control of the kind and amount of wattless current enables control of the alternating voltage at the converter slip rings, thus directly adjusting the delivered direct voltage. It is customary to adjust the converter excitation so that, at light loads, a lagging current is drawn. This current, being small at light loads, does not cause a large voltage drop through the circuit reactance. At about full load the power factor is made unity by increasing the excitation. At overloads a small leading current is drawn, thus causing a rise in terminal voltage. This adjustment of power factor causes the direct voltage to rise with load increase, just as the voltage in the armature of a compound-wound generator rises with increase of

load and maintains the voltage at the brushes approximately constant.

It will be noted that increase of load should be accompanied by increase of excitation in order that the power factor may be brought from lagging values to unity and then to leading values. Increase in excitation with increase of load can be automatically accomplished by the use of a series winding on the converter's poles. This winding is similar to the compound winding on the poles of a direct-current generator, and it serves a like purpose in causing a rising voltage characteristic; but its mode of action, as just set forth, is quite different from that of the simple generator.

Rise in voltage through the passage of leading current through reactance may be accomplished either by large leading currents passing through low reactance or small leading current through greater reactance. The amount of leading or lagging current allowable in a converter is limited by heating of the armature conductors, as previously set forth. It is therefore necessary to have considerable reactance in the transformer circuit. Too much reactance, on the other hand, may lower the overload capacity of the converter. Transformers are now

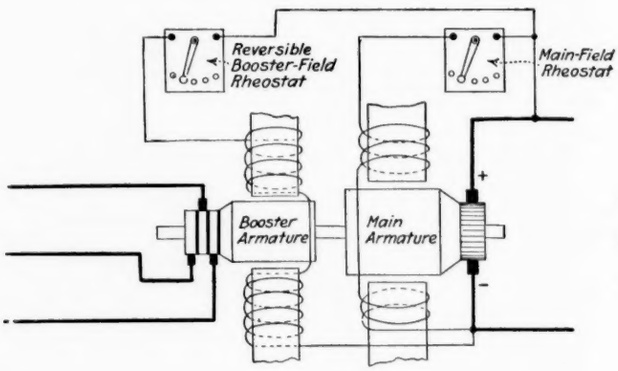


FIG. 24. DIAGRAMMATICAL REPRESENTATION OF A BOOSTER TYPE ROTARY CONVERTER

built having about 15 per cent. reactance, this having been found to be the most effective all-around value. The reactance is secured by arranging the transformer coils to permit a certain amount of magnetic leakage.

The direct voltage regulation of a machine of this type can be controlled by adjusting the strength of the series field, thus influencing the power-factor range corresponding to a given load range. Converters for parallel operation may be so adjusted to secure like regulation curves by shunting the series-field winding. An inductive shunt, wound about an iron core, should be used. The series winding of the converter is inductive; consequently, if a noninductive-resistance shunt is used, the latter will momentarily absorb the current peaks because of the lesser lag therein. The use of an inductive-resistance shunt causes the series winding to assume its relative share of the load even when suddenly applied.

The power-factor range over which a compound-wound converter operates depends upon the excitation of the shunt field. Strong excitation of the shunt field causes increased direct voltage because of the more pronounced leading currents attending overexcitation. It is thus possible to control the direct voltage by adjusting the shunt field, but this should not be done. The shunt field should be used only to locate the power-factor range over which the converter operates. It should be so adjusted that the converter operates at unity power factor at about full load. At fractional loads the power factor will then be slightly lagging, but the decreased work currents permit of this increase in lagging current without overheating. If the direct voltage of the converter is not as desired when the shunt field is adjusted as explained, then the transformer primaries should be connected to different taps to raise or lower the impressed alternating voltage and thus influence the direct voltage to the desired degree. This is an important feature in connection with the operation of converters of this type and one that is often insufficiently emphasized. In order to make easy the proper manipulation of the shunt-field rheostat, it

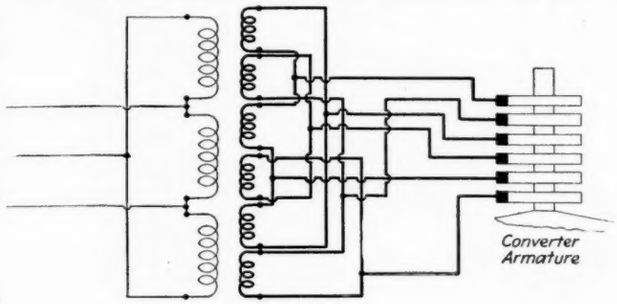


FIG. 25

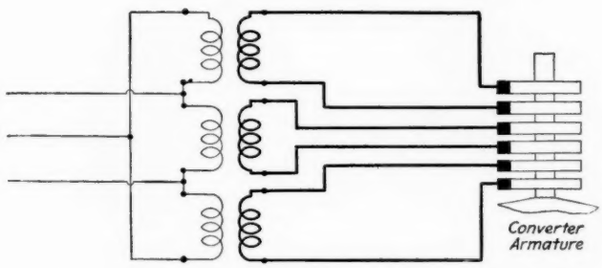


FIG. 26

FIGS. 25 AND 26. TRANSFORMER CONNECTIONS FOR CHANGING 3- INTO 6-PHASE ALTERNATING CURRENT

is desirable that a power-factor meter be included on the switchboard-converter panel.

Inasmuch as the direct voltage of a converter bears a fairly definite relation to the alternating voltage, it follows that, the desired direct voltage being given, a corresponding alternating voltage must be impressed. This voltage usually differs from standard alternating voltages and is obtained through the use of transformers, commonly of the step-down type. Nearly all converters are thus fed through transformers. In order that the exact direct voltage desired may be obtained with correct excitation, it is desirable that the primary windings of such transformers be equipped with taps providing a selection of alternating voltages. If the converter is to be self-starting from the alternating-current end, the transformer secondary windings are provided with partial voltage taps for starting use. The transformer capacity installed for a compound-wound converter is usually equal to the converter's kv.-a. rating and 10 per cent. greater for a booster-type converter. The former type is used more commonly on railway and industrial loads where the peaks, rather than the heating, determine the machine rating. The booster type, used on more steady loads, is more likely to be run at continuous overloads.

It is standard practice to provide a separate transformer bank for individual converter units. This arrangement affords better efficiencies and flexibility and renders parallel operation stable through the cushioning influence of the transformers. Converters operated in parallel from a single transformer bank are apt to be unstable. Single-phase rather than three-phase transformers are almost exclusively used for converter service.

Converters may be operated from single-phase, two-phase, three-phase or six-phase circuits. Six-phase converters are much more common because of economical armature-copper usage, as previously mentioned. The six-phase circuits for the converter may be readily obtained from three-phase feeder lines by proper transformer connections giving 60-deg. rather than 120 deg. phase relations. The secondaries of a three-phase transformer may be connected either in double-delta, as in Fig. 25, or in diametrical connection, as in Fig. 26, to give six-phase service. The diametrical connection is more commonly used since with this arrangement but one secondary coil per transformer is required.

Converters may be started from the alternating-current end through induction-motor action, from the direct-current end as a simple motor or by means of an auxiliary starting motor. Induction-motor self-starting is simplest, cheapest and quickest as synchronizing is not required. It offers the disadvantage of comparatively large starting current at low power factor; however, this method is the most used. Direct-current starting from the busbars or from batteries consumes less starting current, but requires synchronizing and offers greater possibilities for error. Starting with an auxiliary motor also requires synchronizing and is more expensive and complicated. Its only advantage is low starting current. Most converters are now provided with amortisseur or squirrel-cage windings on the field poles, which not only prevent hunting, but render self-starting through induction-motor action quite satisfactory.

Commutating conditions in the rotary converter are normally more favorable than in the ordinary direct-current generator. Owing to the superimposed direct and alternating currents in the armature, the influence of armature reaction is largely neutralized. Commutating poles have been largely introduced in converter design, to furnish a small flux favorable to current reversal. The commutating-pole ampere turns are comparatively small. The use of the commutating pole has made feasible higher speeds and closer ratings in the design of rotary converters.

The commutating poles introduce a tendency toward sparking in case of hunting. This is because of the pulsating-motor current in the armature, not compensated by the commutating-pole ampere turns, yet active in setting up an armature reactive flux across the iron of the interpole face. Effective damping devices to prevent hunting are thus necessary on commutating-pole converters.

Another difficulty with the commutating-pole converter occurs at starting. When alternating current is impressed on the armature for self-starting by induction-motor action, a rotating field is set up. This field cuts the armature coils short-circuited by the brushes and sets up in them considerable voltage, giving rise to heavy short-circuit currents with consequent sparking. As the armature speeds up, the rate of cutting lines decreases and the sparking disappears. This tendency, present in all self-starting converters, is amplified in the commutating-pole machine owing to the presence of the interpole iron, which increases the flux over the short-circuited coils and thus increases the short-circuit currents. In consequence of this tendency commutating-pole converters are usually supplied with a device which lifts the brushes from the commutator during the starting interval. Two brushes are left on the commutator to furnish excitation, but as these are narrow, they do not span many bars and the short-circuit voltage across them is low.

Extracts from a Refrigerating Engineer's Notebook

By H. G. Gibson

SYNOPSIS—Brief descriptions of troubles that an engineer met with in his day's work and how each was corrected.

The ice tank is only a short distance from the brine pump and on about the same level, hence it does not require much pressure to force brine through the tank. A high pump pressure must be carried to get the brine to some of the laboratories in the distant buildings. For that reason the brine entering the ice tank must be throttled to prevent the admission of more brine, under a high pressure, than will flow out by gravity. It has always been difficult to regulate the flow of brine to this tank, and on several occasions the tank has overflowed, spoiling what ice there was in the cans and wasting brine. Finally a bell-ringing alarm was provided so that whenever for any cause the brine in the tank gets above a certain level, a bell will ring in the engine room. The alarm consists of two metallic points (strips of lead) that are the terminals of the bell-ringing circuit; these points are fastened to the frame of the tank in such manner that when the brine rises above a point that has been established as a safe limit, the points become short-circuited through the brine and the alarm bell in the engine room rings.

One evening the ammonia pump developed signs of distress that called for an investigation. The first thing noticed was that the ammonia pump was running slower than usual, the throttle valve having the customary opening. The pump was given more steam, but this only brought it up to speed again for a short while after which it slowed down again. The throttle valve was opened wider and wider until it was nearly fully open, but even then the pump would not run fast enough to do its customary work. It acted as though the exhaust steam passages leading from the pump had become partly

closed, making the pump work against high back pressure. Examination of all the valves on the exhaust side of the pump did not disclose any out of the ordinary condition, and it was a question what to try next.

Going back to the pump, the operator opened the throttle wide and waited to see what would happen. The pump speeded up a little and then slowed down again. A hissing sound from the direction of the exchanger then attracted the operator's attention, and he looked over that way. One look was enough. He saw streams of ammonia liquor spurting out around the top joint of the exchanger at every stroke of the pump. Apparently the trouble was in the ammonia side, and later events proved that to be the case. A piece of waste, probably left in the machine when it was overhauled, had worked its way through the pipe connecting the exchanger and analyzer, Fig. 1, until it came to the valve *A*, where it lodged. When the pipe and valve were taken down for examination, the waste was found wedged in the valve chamber.

Since the foregoing incident, computation has shown that there must have been between 700 and 800 lb. pressure per square inch on the exchanger and the pipe

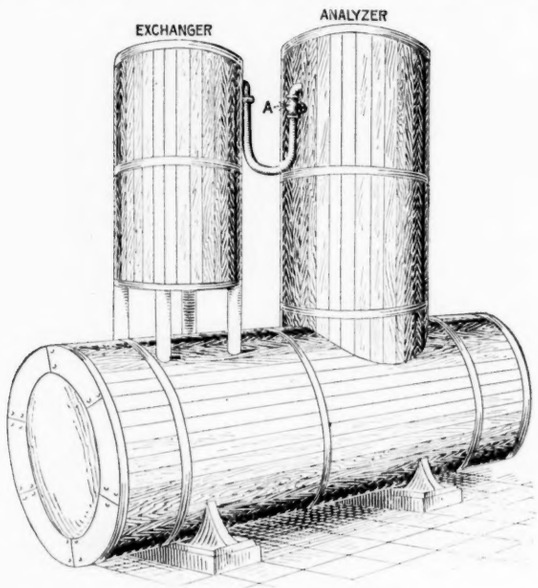


FIG. 1. SHOWING LOCATION OF LEAK ON ANALYZER CONNECTION

connecting the exchanger and analyzer, when the top head of the exchanger started to leak. No wonder the header bolts stretched!

As stated before, some of the laboratories to which the brine is piped are situated in buildings several hundred feet from the ice machine. The building housing the power plant is connected by tunnels with the laboratory buildings, and the brine-circulating pipes are run through these tunnels. One day a brine leak developed about midway between the ends of the tunnel leading to the east building. This seemed rather strange, because both building and tunnel had just been completed, and the brine had never been used there except to test out the piping for leaks. The pipe work was put up by reliable men, and the final test proved that every joint was tight. No brine had been circulated in that building since the final test, about a week before, so why the sudden leak in the tunnel?

That question went unanswered until it was discovered that the valves on both the supply and return brine pipes to the east building had been shut off at the plant end of the tunnel at the conclusion of the test. The brine was below the freezing point when the test was conducted. When the valves were shut at the conclusion of the test, it is plain that the supply and return pipes to that building were full of brine; and if the return valve were shut first, the brine contained in the pipes would be under pressure. It is evident that under these conditions the cold brine bottled up in the pipes had no space in which to expand as it warmed up to tunnel temperature, and the consequence was a burst pipe.

Sometimes, in taking foreign gas out of a machine, there may be a question as to whether there is enough coming out to justify one in leaving the gas valve open.

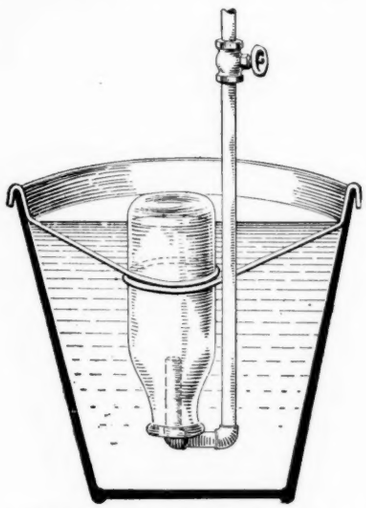


FIG. 2. ARRANGEMENT OF WATER BUCKET AND BOTTLE FOR INDICATING ESCAPE OF AMMONIA

When such a question arises, take a pint or quart bottle with a large mouth (a milk bottle is handy for this purpose) and fill it with water. With the open end of the bottle submerged in a bucket of water invert it, and hold it in this position; and at the same time insert the end of the foreign gas hose in the end of the inverted bottle, Fig. 2. Some idea of the amount of air and noncondensable gases coming out may be obtained by taking note of the rapidity with which the bottle empties of water as it is displaced by air or gas. Ammonia gas would of course be absorbed by the water. [One must not forget that time is required for such absorption.—Editor.]

The generator gage glass had been acting peculiarly, and we were not certain that it was showing the true level of liquor in the generator. On account of a repair job in the steam line to the ice plant that necessitated shutting down the machine, we had a chance to examine the glass and valves. The trouble was finally located in the bottom gage-glass connection. The rubber ring packing that had been used to pack the gage-glass stuffing boxes was too soft for the purpose, and as the nut was screwed down to prevent leaks, the soft rubber

squeezed under the end of the glass and stopped up the passage. When the glass was repacked a hard rubber ring was placed in the bottom to prevent a recurrenc of the trouble.

The gage-glass valve stems on the condenser were also packed at this time. Heretofore a twist packing had been used for this purpose, but such packing has the disadvantage of drying out quickly and becoming wedged into the clearance space between the stuffing box and the stem. For this reason one of the boxes was packed with rubber ring packing, but it would not do and had to be taken out again. The rubber ring packing is all right for gage-glasses but it binds so tightly around valve stems that it is next to impossible to turn the valve. We had to go back to the old "twist" for valve stems.

Some time ago the generator was thoroughly overhauled. Both heads were off and new gaskets were put in when the heads were replaced. Needless to say, the header bolts were made as tight as it was possible to make them, and we were of the opinion that they were tight for good. The generator ends were then covered up, magnesia packed around them, and they were forgotten.

About a year after the heads were covered a leak developed somewhere around the plant, and though it produced quite a "smell," it could not be located. Finally one of the hunters put his nose to a joint in the jacketing near one end of the generator and got a "shock" that sent him reeling backward. That meant that the covering would have to come off to get at the header bolts. On taking off the cover some of the nuts were found to be only hand tight. The header bolts on both ends of the generator were found to be in the same condition.

It is more of a job to take off and replace the cover in a task of this nature than it is to tighten up the bolts. Also the removal and replacing of the magnesia around the joint necessitates making more or less of a dirty mess each time the operation is performed. The covering is still off the generator ends. When it is replaced next time an effort will be made to devise a way of putting it on so that it may be easily removed. The generator jacketing should be so constructed that the removal and replacement of the part that covers the header bolts would be a matter of only a few minutes work. Then these bolts could be inspected at frequent intervals and tightened whenever necessary.

Induction Relays for Selective Overload Protection

The General Electric Co. has recently perfected an induction-type time-limit overload relay, shown in Fig. 1, which is particularly applicable to those systems where extreme accuracy in timing is required for tripping two or more air or oil circuit-breakers selectively.

The operating or characteristic curves for the various time-current settings are entirely separate and distinct at even the heaviest overloads and never become instantaneous. This is because of the inherent characteristics of the relay, which produce a curve consisting of an inverse time portion up to approximately 2,000 per cent. of minimum-contact closing current, blended into a definite time portion with a slight downward slope. Consequently the relay will do the work ordinarily required of both inverse and definite-time-limit relays. The heaviest overloads do not disturb the form of the curve nor cause vibration or chattering of the moving parts.

The relay is made in single-pole elements, is circuit-closing, and operates with a time delay which is inverse for the lower current values and which approaches a definite minimum for the higher current values. It is designed for use in the secondary of current transformers, the normal load rating being five amperes. However, by means of the current tap plate, Fig. 2, the relay may be

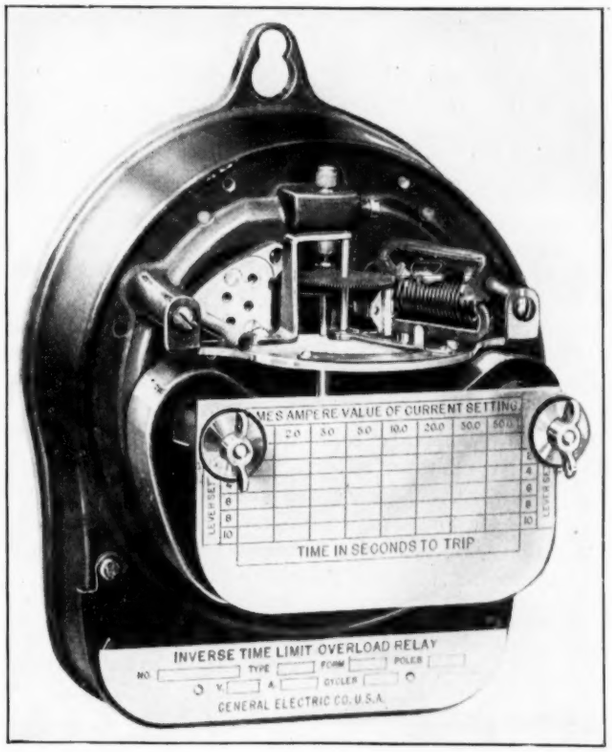


FIG. 1. INVERSE-TIME-LIMIT OVERLOAD RELAY

set for 4, 5, 6, 8 and 10 amperes, positive operation being obtained throughout this range.

The contacts are closed on overload by the rotation of a disk actuated by a U-shaped driving magnet with shading coils on the polepieces. No tripping current is carried through the revolving parts. When the contacts have been closed, they are firmly held in that position until tripping occurs, by the armature of a holding coil connected in series with the contacts, the trip coil of the air- or oil-break circuit-breaker and an auxiliary switch which opens when the breaker is tripped. This insures current on the trip coil continuously until the circuit-breaker opens, and prevents flashing at the relay contacts.

The values given in vertical columns 1 to 10 on the index plate, Fig. 2, are the time delays that will be obtained at the different degrees of overload represented in the times-current-tap-setting columns at the extreme right and left sides of the index plate. The factors appearing in the times-current-tap-setting columns, when multiplied by the current tap setting, represent actual secondary-current values.

To obtain a given current setting, it is only necessary to insert a tap plug in the current-tap plate at the de-

sired current marking. Time settings are obtained by moving a small lever in the front and near the top of the relay, to the column in which the desired time appears. For example: Using current tap 6, it is desired that the relay be set to operate in 1.20 sec. with 120 amp. in the current-transformer secondary; 120 amp. is 20 times current tap setting 6. Referring to Fig. 2 in the horizontal row of the time values opposite factor 20, is found

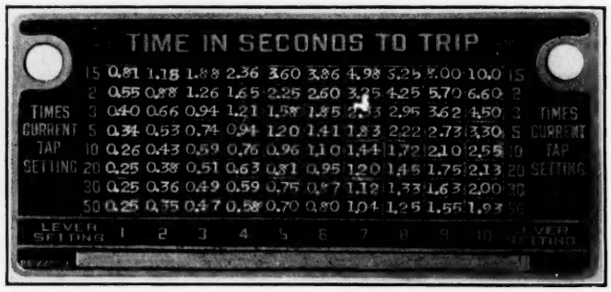


FIG. 2. INDEX PLATE FOR RELAY CURRENT SETTINGS

1.20 sec., the time desired, in vertical column 7. Therefore, in setting the time lever, the index point will be placed exactly over graduation mark 7 on the lever scale.

If the nature of the work is such as to require time settings intermediate to those given on the index plate, the proper time-lever setting may be closely estimated or, if necessary, may be calculated by interpolation. For a given time-lever setting, the characteristic curves for all the current-tap settings are practically identical.

Curves for consecutive time-lever settings for less than maximum time fall below the maximum curve shown in Fig. 3, in the same order as the settings.

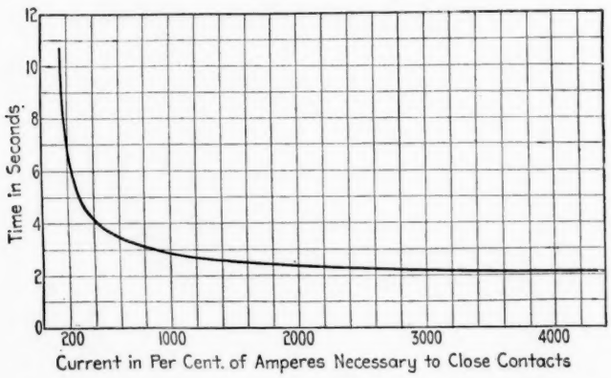


FIG. 3. CHARACTERISTIC CURVE OF RELAY AT MAXIMUM TIME LEVER SETTING

The contacts are designed for use on direct current. They are made of a special metal which is noncorrosive and which has high heat resistance. They will carry 18 amp. momentarily without damage to the contact surfaces. The current-closing capacity is therefore ample to trip even the largest circuit-breakers.

A thermostatic device within the relay compensates for any variations in room temperature. Following wattmeter construction, the movable element is supported by a jewel bearing, thus minimizing friction and eliminating vibration. A dust-proof glass cover protects the operating parts of the relay and facilitates the reading of the index plate.

The relay is built in two styles, one for 25- and one for 60-cycle circuits. Although the principle of operation and the inherent characteristics are the same in both relays, they differ only in slight details of construction.

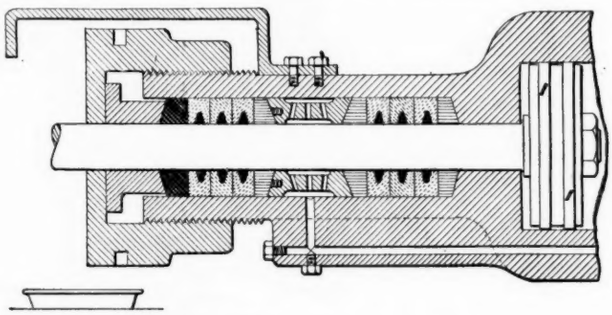
Life of Ammonia Pump Rods

By D. L. Fagnan

As is generally known, the life of the piston rods of ammonia pumps is comparatively short, and the rods must be packed quite frequently to prevent leakage of ammonia. This frequent packing wears the rod considerably, and many engineers keep on hand several sizes of packing rings, so that as the rod wears smaller the packing rings used are correspondingly smaller, to make a tight joint. Keeping on hand various sizes of packing rings and repacking the rod every two or three weeks is a source of loss of ammonia.

I have had considerable experience as a troubleman in refrigeration work, which has brought me in contact with many operators and with many plants. Inquiries of 230 operators in one year showed that 25 had to renew the rod every three months and had to repack it every three weeks. This was in 24-hour service, although most of the plants running 18 hours experienced the same trouble. Correspondence with various ammonia-pump manufacturers regarding the extreme wear of the rod corroborated the experience of the operators.

I have done considerable experimenting with materials for pump rods, and that which has given the best service is a high-grade tool steel. A local machinist made



STUFFING-BOX FOR COMPRESSOR

one of these rods without annealing it. This rod was packed with fibrous packing having a metal or babbitt insertion, the stuffing-box being packed as shown in the illustration.

This rod has been in service for more than a year, running 18 hours a day in summer, and 6 to 8 hours a day in winter, and the wear is only 0.002 in. The gland has been taken up approximately $\frac{5}{16}$ in. in all this time, and the rod is operated at about blood heat. From 5 to 6 drops of oil per minute is fed over the rod, and it is only when starting that one gets a slight odor of ammonia, say until after the second or third stroke is made. From then on the odor of ammonia is not noticeable.

In packing the rod it is advisable to put a standard ammonia packing rings in the bottom of the box and then alternate with the metallic rings, or rather the fibrous rings having metallic insertions. The last ring to be put on should be of a good resilient rubber, placed next to the gland where it can take up expansion and contraction.

To Determine the Specific Heat of a Gas at constant pressure, the gas is led through a worm, heated to a normal temperature, then passed through a spirally divided chamber immersed in water in a calorimeter. The specific heat can be calculated if the resulting rise in temperature is known.

Visits of Inspector Brown--XX

By J. E. Terman

SYNOPSIS—Brown works industriously on a table giving the spacing of stay-bolts under the code formula. He speaks to the Chief about short-cut methods of doing the work and is given some pointers regarding plotting formulas. These include that of the pressure exerted by a column of water for various heights and also diagramming the multiplication tables.

Brown had been working industriously on a table giving the spacing of stay-bolts under the Code formula, since the table in the Code did not fully meet the requirements of this inspection work. He found it a laborious task to make the table as complete as he desired it, by working out each result by the formula, and he thought that possibly there might be a short cut in doing the work, and so decided that he would speak to the Chief about the matter.

"Chief," said Brown upon entering the office, "I want to make up a table to show the proper spacing for stay-bolts under the Code formula, for I find that the table in the Code is arranged only for plate thicknesses progressing by even sixteenths of an inch, while I often come across furnace plates that measure in thirty-seconds of an inch, and to be certain of the maximum pressure that I can allow, I have to figure each case out by the formula. I have been trying to work up a more complete table, but it is a tedious procedure and I thought that you might tell me some way that would make the task easier."

"Well, Brown, I can tell you right away that if you had bought that slide rule I advised you to get some time ago, it would lighten up your work tremendously, as you no doubt would have been able to use the rule with confidence by this time," said the Chief. "There are several ways by which the labor of making a table that would represent the maximum spacing for stay-bolts under the Code rules might be much reduced over the 'main-strength-and-awkwardness' method that you have been employing. The way to get up such a table with the least expenditure of energy is by plotting the formula given in the Code, and there are different methods by which such a diagram might be made, but if we take up the matter we want to start at the foundation.

"The subject of plotting formulas is so important that I think you might profitably spend a lot of time in learning the principles involved. I could, of course, in a short time show you one method of making a diagram to represent the values for stay-bolt spacing under the Code formula, but you would not understand the reasons involved and could not use the method again for solving similar problems that are continually arising in your inspection work. If you will come in every afternoon that you have an opportunity, I will try to explain the principles of plotting formulas, so that you may make use of this short cut."

"I will certainly be glad to do that, Chief. My work is not so heavy now, and by getting an early start on the light days, I can usually spend a good portion of the afternoon with you."

"Well, Brown, since you evidently have some time on your hands, we will have the first lesson now. You have often seen diagrams made up of curved lines, but we will not be interested in this class, for as a rule, numerous points have to be calculated in order to draw the required curves, and the amount of labor involved in making such diagrams is often as great or greater than that required to get up a table of values such as you have been attempting to get up for stay-bolt spacing. We will confine our attention to diagrams that may be made up of straight lines, where it is necessary only to obtain the location of two points on any line in order to draw it, and all the intermediate values or those beyond that are within the range of the diagram may be read off without further calculation.

"Suppose, Brown, that you desired to make up a table to show the pressures due to the weights of columns of water of different heights. Then, since a cubic foot of water weighs 62.5 lb.; you would know that a column $\frac{1,728}{12} = 144$ in. high would exert a pressure of 62.5 lb. per sq.in., or that a column 1 ft. high would exert a pressure of $\frac{62.5}{144} = 0.434$ lb. per sq.in. Therefore, multiplying the height of a column of water in feet by 0.434 would always give the pressure per square inch exerted

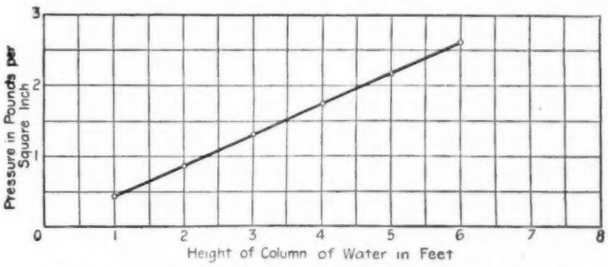


FIG. 1. DIAGRAM OF THE WEIGHT OF A COLUMN OF WATER

by the weight of the water in the column. Saying the same thing in other words, the weight in pounds of a column of water 1-in. square is its height in feet multiplied by 0.434. If you wanted to plot these results on a diagram, you could draw horizontal lines to represent pressures in pounds per square inch and vertical lines to represent heights in feet" (showing Fig. 1). "On the vertical line indicating 1 ft. height there would be indicated a point at 0.434 lb. above the base line and on vertical line 2, a point would be at $2 \times 0.434 = 0.868$ lb. above the base line and on line 3 a point $3 \times 0.434 = 1.302$ lb. above the base line, and so on as far as the diagram should be extended. You will see, Brown, that if such a diagram is accurately constructed, all the points obtained are on a straight line, and if we had been aware of this fact at the start, it would have been necessary only to find the location of the point on the first and the last vertical line of the diagram and then draw a straight line through these points across the diagram, and the points where this line crossed the other verticals would indicate the pressures exerted by the columns of water of the height denoted by the respective vertical lines. It happens in this case that the straight line

across the diagram passes through the intersection of the vertical and horizontal lines at the starting point or zero, so that it would only have been necessary to obtain one point, say on the vertical line No. 6, and connect this with zero, in order to complete the diagram and to be able to tell what pressure per square inch would be exerted by a column of water of any height within the range of the diagram."

"Say, Chief, would any other multiplier besides 0.434 produce straight lines that would pass through zero?"

"Yes, Brown, if the pressure exerted by a column of water 1 ft. in height had been 5 or 6 lb., or 600 lb. per sq.in., the pressure exerted by a column of water of any height would be indicated by the intersection of the horizontal lines with a line drawn through zero and a point on one of the vertical lines which would represent the pressure for the particular height of column represented by the vertical selected."

"Well, Chief, if that is so, why would it not be an easy matter to make up the multiplication table in that way?" said Brown.

"Sure," answered the Chief. "The multiplication table may be diagrammed with ease, and since one is familiar

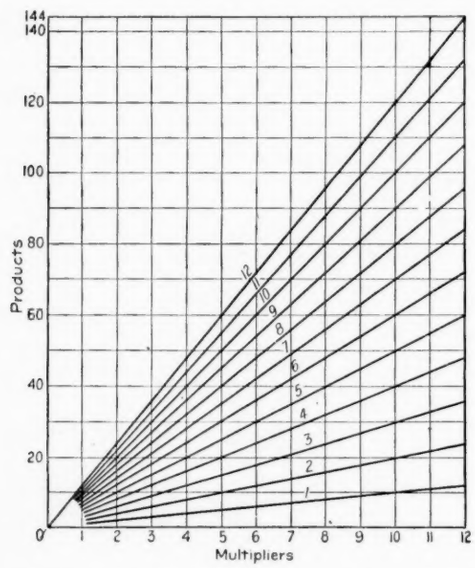


FIG. 2. DIAGRAM OF THE MULTIPLICATION TABLE

with the multiplication table, diagramming it may give a good idea of how this method may be used to save time and labor, as well as increase the usefulness of the information obtained. Here is a piece of cross-section paper" (producing Fig. 2). "We will assume that the horizontal lines represent products and the vertical lines multipliers; diagonal lines will be used to represent the different multiplicands. As I told you, these diagonal lines will all pass through zero, and therefore it is only necessary to find one other point on each of the lines to be able to locate it. Of course, to make the diagram accurate, it is best to have the point used to locate each line near the opposite side of the sheet from the vertical line passing through zero, and for the sake of simplicity, we will use 10 as a multiplier, and the points of intersection of the diagonal lines representing the multiplicands 1, 2, 3 and so on, with this vertical representing the multiplier 10, will be at the intersections of the horizontal lines representing products of 10, 20, 30 and so on; that is, 10 times the multiplicands. Connecting these points with zero, we have the diagram completed. The time taken to draw this diagram was less than would be necessary for you to write the multiplication table, although you are perfectly familiar with the values to be put down. Although the drawing of this diagram was such a simple thing, it will pay to look at it carefully, to see just what we have. It is easy to see that it will give you any answer that the multiplication table will give, and it is capable of doing more than this.

"You understand, Brown, that division is simply the reverse of multiplication and this diagram may be used to divide numbers as well as to multiply them. For example if you wish to divide 72 by 6, follow the horizontal line representing the product 72 across the sheet to the diagonal line representing 6, and it is seen that this intersection is on the vertical line representing 12, the answer. Another thing about this diagrammatic representation of the multiplication table is that the lines may be used to represent the numbers placed on them or any numbers obtained by multiplying such numbers by 10. For example, the diagonal line marked 7 could represent 0.7, 7, 70, 700, 7,000 and so on, and if the paper was large enough, the diagram could be extended indefinitely, and of course the larger it was made the more closely it could be read. Such a diagram as we have been discussing, Brown, as far as multiplication and division was concerned, would do the same work for you as a slide rule, but it would not be so handy to use. In using the different values that may be assigned to the lines, it would be necessary to keep track of where the decimal point should be located. You will observe that the increases on the vertical line representing the multiplier 10 (and the same is true for the intersections on any other vertical) are equal to one another, and this is so because the increases in the multiplicands are uniform; that is, one. I will call your attention to this matter again when we get deeper into the subject of digramming formulas, for it often tremendously reduces the labor involved in making such diagrams. You have now only enough time left to finish your report writing, and we will have to put off further discussion until another afternoon."

"Well, Chief, I am certainly obliged to you for the instruction that I have received today, and it seems possible that before we get through with the subject of diagrams, I may be able to make one that will take the place of the slide rule you suggested and save the price for something else."

Casehardening with Cyanide—For many purposes a thin, hardened outer surface is required on wrought iron or steel articles. This can be quickly secured after finishing by the use of potassium cyanide. All iron and steel, roughly speaking, will absorb carbon when at a temperature of from 800 to 850 deg. C., or, in smithy terms, from bright to cherry red. This is taken advantage of in surface hardening, the metal being heated to this point, preferably in some form of muffle, even rough. At this stage the metal can be plunged into a bath of melted cyanide or rubbed with a lump of cyanide held in a pair of tongs until the metal becomes dull red, after which reheating to the hardening temperature and immersing immediately in cold water to secure the greatest hardness possible. Properly done, this should turn a file. A better plan, where the size and shape of the articles permit, is to coat a piece of leather with powdered cyanide and press the heated articles on this, repeating the heating and pressing several times, afterward reheating and quenching. For deep casehardening the usual method of packing and heating for several hours must be adopted.—"The Practical Engineer, London."

Editorials

Inertia and Momentum Effects in Reciprocating Engines

It is not always the visible and evident with which the engineer has to deal, and many effects, both beneficial and disturbing, cannot be understood by the consideration of the tangible alone, but must comprise also the evidence or substance of things not seen.

The effect of steam pressure upon the piston of an engine is readily understood, although not everybody who works with steam realizes that it is the continued bombardment of the surfaces by myriads of molecules moving at inconceivable velocities which produces that pressure. But whether or not he realizes that it is the effect of overcoming momentum, the steam engineer is accustomed to study pressure upon the indicator diagram, to measure its effort upon the crankpin and to consider its action, in the form of compression, in helping to bring the parts to rest.

There is, however, a less evident force than that exerted by the steam, which makes the actual distribution of pressure upon the crankpin very different from what it appears to be from the indicator diagram; namely, that introduced by the inertia and momentum of the reciprocating parts. Even with engines of moderate speed this has an important effect in reducing the initial effort upon the pin and in magnifying that effort after it would appear on the indicator diagram to have been reduced by expansion to insignificance. With high-speed engines it may have the effect of completely reversing the apparent distribution of pressure. While the indicator diagram may show a high pressure at the commencement of the stroke, petering out to little or nothing at the end, a study of the inertia effects may show that, owing to the absorption of energy in getting the parts into motion, there is little effort left to be exerted upon the crank in the early part of the stroke, while the return of this energy by the parts as they are brought to rest brings the effectual pressure on the crankpin at the end of the stroke above what it was at the beginning. An example of this will be found in the diagrams reproduced on page 678.

Charles T. Porter, during the brief experience in the practice of law with which he commenced his career, became interested in a marble-cutting machine. The difficulties inherent in the reciprocation of a gang of heavy saws at high speeds led him to apply his analytical mind to the forces involved in reciprocating bodies, and when later he became associated with John F. Allen in the evolution of the Porter-Allen engine, which ran like a sewing machine at unheard-of speeds, he brought to the task the clearest comprehension of the forces involved and their mode of action that had ever been directed to this feature of engine design. The exposition of the subject in his book upon the Richard's Steam Engine Indicator is a classic, and the principles discussed and processes employed should be more familiar to and more frequently employed by users of that instrument. An attempt to show them in all their simplicity, to make their employment easier by the use of tables and to show their application to typical cases is made in the article, "Compression from a Mechanical Point of View," on page 674, *et seq.* *Power* readers who use the indicator will find this analysis very easy to make and very illuminating, especially in the case of high-speed engines. The inertia effects, and therefore the necessity for compression, increase as the square of the number of revolutions per minute, but only directly as the stroke, so that an engine running at twice the number of revolutions as another having twice the stroke would require four times the compression for the same weight of reciprocating parts, notwithstanding the piston speed would be the same in both cases.

Frequency Standardization

The tremendous progress made by the electrical industry since its beginning about 1878 has necessarily made the development of rather an erratic nature. No industry could have become the most important factor in our industrial and economic existence in so short a time unless a great number of workers had turned their attention to its development. Naturally, among this large number of engineers and inventors a great diversity of opinion existed. This at once becomes manifest when consideration is given to the different methods of transmission and distribution of electric energy for light and power. The methods used in alternating current are even more diversified than in direct-current systems. Whereas in the direct-current system the diversity can be only in voltage, with alternating current it is volts, frequency and phases.

The first alternating-current systems operated at 133 or 125 cycles single-phase. This frequency seemed to be the correct practice for the generators and transformers used at that time, as this type of apparatus was less expensive to construct than the lower-frequency equipment. In some cases 116 cycles were used.

With the development of the polyphase system it was found desirable to use a lower frequency, consequently the sixty-cycle system was adopted. As the transmission voltage was increased, it was again found desirable to decrease the frequency and 25 cycles found its place in the category of alternating-current systems. This frequency had several advantages, among them low charging current in high-voltage transmission lines, its adaptability for synchronous converters and series railway motors. In the case of the synchronous converter, however, this advantage has almost disappeared, as in late years the sixty-cycle converter has become a successful machine.

For various reasons there have been a number of other frequencies adopted, such as 66.6, 62.5, 50, 40, 38 and 30 cycles. These are some that we find in our own country. Turning attention to Europe, we find in addition such frequencies as 100, 83, 42, 33 and even 15 cycles, the last being used for railway work.

When it is taken into consideration that many of these systems operating at one frequency cannot be tied into other systems operating at a different frequency and that,

when it is possible, expensive and inefficient machines are required, the necessity for standardizing frequencies at once becomes apparent. To tie a 25-cycle system into a 60-cycle the only commercial set available is a ten-pole twenty-five-cycle machine connected to a twenty-four-pole sixty-cycle, which gives a speed of 300 r.p.m. This speed is about one-fifth of what it is possible to economically design even very large machines for, consequently the cost of this set will be far in excess of what it would be if the most economical design could be taken advantage of.

The question what should be the standard frequency is one that is not definitely settled. In the western part of the country preference is given to the sixty-cycle system; in the East the twenty-five-cycle is looked upon as preferable. Outside of the fact that twenty-five-cycle systems have been installed to a very large extent and had various advantages at the time they were adopted, there exists at the present time no reason for their continuance except for single-phase railway work. In almost every other case the sixty-cycle system has distinctive advantages over systems operating at any other frequency. Therefore, it seems that, when the industry is relieved of this conglomeration of frequencies by the adoption of some standard, the day will be hastened when electricity will be used universally for the transmission of energy.

The Meaning of Curves

Inspector Brown is drawing from the Chief a series of talks which will interest many others than boiler inspectors. In his last two talks he got onto the subject of diagrams or plots, and in those which are coming he shows very simply and understandably how such diagrams are plotted and gets one into coördinate axes and curve formulas without his suspecting what is being done to him.

As to the comparative merits of tables and diagrams—they are not comparable. For giving precise values for a limited number of conditions the table is simpler, better and more convenient and satisfactory. If one is content with approximate values, a diagram gives him an infinite number of them on a single page. The entire book of steam tables could be represented upon a single chart, but the chart would have to be unreasonably large to give the precision of the tables or to be legible to the same number of decimal places.

It is in showing the relations of matters, the effect upon one factor of a change in another, the distribution and trend of things that the diagram is principally useful. In doing this it will frequently furnish a satisfactory numerical value. A diagram which anyone can plot for himself of the relations between the pressure and temperature of saturated steam would be very illuminating to the one who made it, help him to appreciate as he never did before how those factors varied in relation one to the other and to fix in his mind the corresponding values at the principal points, but he would probably continue to use the steam tables for precise calculations.

The temperature-entropy diagram throws daylight into heat-engine problems as nothing else can. A Mollier diagram in which the heat contents of steam are plotted against its entropy shows at a glance what the quality of the steam will be after expansion from a given initial to a given final condition, how many B.t.u. per pound are converted into energy or what velocity would be generated in the mass, as through a nozzle, by such expansion. Even if greater accuracy than that to which the chart can be read is essential for final results the chart is better than a mass of calculations in determining the effect of changing conditions.

The columns of *Power* have been replete with diagrams and charts, notably those contributed a few years back by N. A. Carle, which are much valued by designers and engineers as of use in blazing out a design if not in the determination of actual final values; and such diagrams applied to actual calculations often point out errors which might not otherwise have been detected. Numerous tables of riveted joints have been prepared. A few diagrams will cover the whole subject and show the designer instanter what efficiency is possible under given conditions or what he must do or have to obtain a joint of a given efficiency, and a collection of such diagrams will soon be available to the designer of riveted work.

A Practical Safety Movement

The stuff a man is made of or, more properly, his physical condition is being recognized more and more as a deciding element in the employment of help of all kinds. In a great number of known instances, accidents have been caused and even lives lost by reason of a physical defect in the makeup of someone, perhaps not even directly concerned in the work being done—a defect the existence of which was unknown and unsuspected by any one, even the subject himself. No doubt many more accidents than can be definitely known are the direct result of such unfitness for the occupation chosen.

A number of the more progressive business organizations throughout the country require that every man employed be physically fit and that every employee be subjected to a thorough medical examination at frequent intervals during his period of employment, so as not only to avoid as far as possible the transmission of disease, but also to guard against assigning a man to duties he is not bodily or tempermentally fitted to perform. The benefits derived from such a process of elimination by far outbalance the comparatively few instances in which an apparent hardship is worked. It certainly is a means of doing the greatest good to the greatest number of people.

Among those active in this work may be mentioned the National Founders Association, the National Association of Manufacturers and the National Metal Trades Association, working through their joint conference board for the mutual good of all. Protective devices and methods are also developed and given an impartial tryout. It is therefore possible to carry on the broader work of safety in a systematic manner.

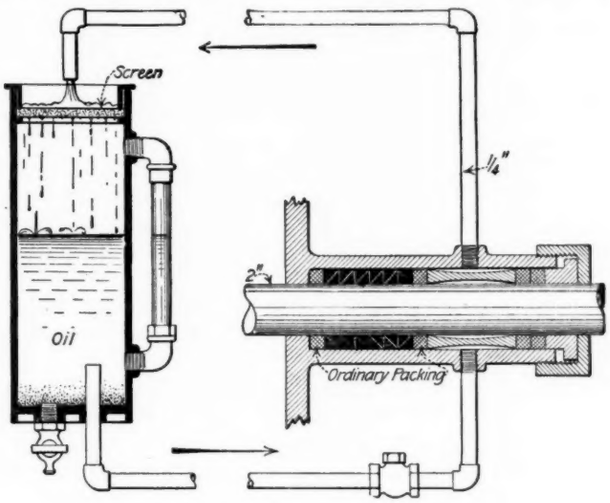
It must not be understood to be an altruistic movement dictated by an awakened conscience, for whatever else it may be, it is a paying proposition and is founded on a firm business basis supported by statistics gathered in the course of recent years in all the more important manufacturing enterprises.

It is certainly within the reasonable rights of an employer to safeguard his business interests to this extent without opposition or prejudice. The cost of "breaking in" a new man rests on the employer and is like the installation of a new machine, which cannot be done without cost if the machine is finally rejected.

Correspondence

Oiling Ammonia-Compressor Rod

The illustration shows the way I have kept the piston rod oiled on an ammonia compressor for the last three years. The oil reservoir is set to one side of the lantern,



AMMONIA STUFFING-BOX OIL RESERVOIR

with the bottom of the sight glass level with the bottom of the rod. The oil will then circulate from the bottom connection of the reservoir, through the lantern and back to the reservoir by way of the upper or discharge pipe. This is a whole lot better and less trouble than the old-style pump run from the shaft by a belt.

Appleton, Wis. ALBERT WARNER.

Coal and Ash Handling at Grundy Plant

In the comparative operating costs published in H. J. Edsall's article in the issue of Oct. 3, page 480, an important item of expense has been omitted in the cost of operating the coal- and ash-handling machinery. The power costs, which are by no means negligible, are not taken into account.

The total lift for the coal (scaled from the layout on the basis of other dimensions given in the article) is about 72 ft. and that for the ashes, 67 ft. As 200,000 lb. of coal is elevated daily, and assuming 13 per cent. for the weight of refuse removed from the ash-pits, the total weight of ash and refuse elevated is 26,000 lb. per day.

The theoretical equivalent kilowatt-hours per 24 hr. required to elevate the coal would be $(200{,}000 \times 72 \times 0.746) \div 33{,}000 = 325.52$, and for the ashes, $(26{,}000 \times 67 \times 0.746) \div 33{,}000 = 39.37$, giving a total theoretical power consumption of 365 kw.-hr. per 24 hr. To this figure must be added the power for the horizontal carry, for the crusher and for the traveling weigher hopper. When the friction loads are added and motor efficiencies and line drops are taken into account, we can safely assume a daily load of 600 kw.-hr. at the switchboard.

If the cost per kilowatt-hour is as low as 0.5c. for this plant, the annual power charges to the boiler room for coal and ash handling would be $600 \times 365 \times \$0.005 = \$1{,}095$. When taken into account this item would make a considerable reduction in the savings presented by Mr. Edsall.

Although in this particular case the saving is such as to warrant the expenditure of the additional $15,000, such an omission would, in a great many instances, bring about changes that otherwise would not be made.

Copperhill, Tenn. L. V. CURRAN.

Horizontal vs. Vertical Diesel

A few years ago, after completing a series of tests with oil engines of different makes, an engineering concern engaged me for three months to find out by means of two competitive designs, if such opinions as voiced by H. R. Wass on page 530 had a more solid foundation than mere assumption. My sympathies were from the beginning with the horizontal type, but the facts disclosed by our preliminary search were mostly in favor of the vertical engine in the case of larger capacities.

The first point the design brought out was that the frame of the vertical type could be made considerably lighter than that of the horizontal engine, as the bending moment in the frame of the latter could easily be eliminated in the vertical. This is of great importance where the principle involves high cylinder pressures, and explains why builders of the Diesel and similar types adhere to the vertical type.

The surface space required by the vertical type turned out to be only three-fifths that of its competitor, calling for a smaller foundation, and as the forces were of vertical nature only, it could be much lighter.

Again, the shape of the vertical cylinder head was superior in many points, and the dead volume was to the advantage of the vertical type—canals and lost corners cannot be advantageous to proper combustion.

The number of cylinders is practically unlimited. Higher speeds with resulting smaller engines and better balancing are points of importance. The mechanical-efficiency factor of vertical engines is greater than that of the horizontal type, as the friction and wear of the piston and crosshead are eliminated, cylinder lubrication is better and piston leakage is reduced considerably by the free movement of the rings. The unobstructed expansion of the vertical type in many cases explains why there are more failures of horizontal than of vertical engines.

Many readers might appreciate a brief "bill of particulars" as to bore, stroke, valve diameter and lift, number of revolutions and fuel used, to know what outworldly or antiquated Diesel compels Mr. Wass to delegate a small army to remove the doors, or requires four

men to spend sixteen hours in replacing an innocent piston ring. There is an impression that the accessibility to pistons, valves, gearing and pipes was the recognized feature of the vertical types. I remember a few unsophis ticated Lascar coolies doing a similar job before a belated breakfast. JAN SPAANDER.

Brooklyn, N. Y.

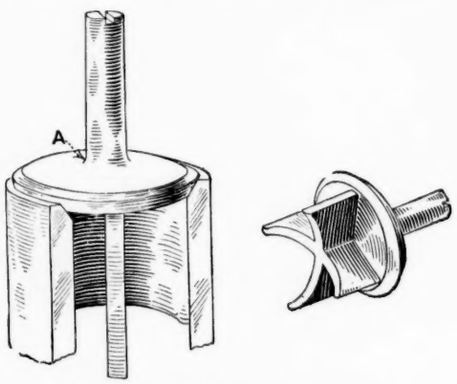
Beveled Seats for Pump Valves

In the issue of Oct. 17, page 566, George Clevenstine asks regarding beveling the seats of feed-pump valves. I would suggest that there is considerable expense involved in this change and that it probably will be less expensive and entirely satisfactory to equip the pump with new seats, if they are cut too badly to be dressed, and valves of rubber of a suitable hardness for the temperature of the water the pump has to handle, as recommended by any reliable maker or dealer in such valves. If Mr. Clevenstine will do this, his troubles will then be over except for an occasional renewal of a valve or seat at a slight expenditure of time and money.

I consider metal valves a nuisance on account of the noise they make, the time and trouble required to regrind them and the fact that they don't "stay put" for any great length of time. I have changed several old pumps that had metal valves and had no further trouble with them. JOHN M. COLEMAN.

Somerville, Mass.

As to which is the better, beveled or flat seats for pump valves, my opinion is that it depends a great deal on the kind of service required of the pump, the sort of liquid



STEMS SOMETIMES BREAK AT A

to be pumped and whether it is a high- or low-pressure service.

We use ten high-pressure feed pumps, all equipped with beveled seats and disks. These valves, in so far as the design of the seats is concerned, are satisfactory, remaining tight in daily use under pressures as high as 500 lb. There is very little trouble from cutting or wear, and when it does become necessary to regrind them, it is an easy matter. Trouble is sometimes caused by the stems breaking off at point *A* in the illustration, but this would be avoided if the valves were designed as shown in the illustration by Mr. Clevenstine. It is certain that the flow of water would be easier over a 45-deg. angle than it would be over a 90-deg. one. Flat valve seats have the disagreeable habit of collecting particles on the seat, keeping the valve raised. This seldom occurs if the seats are beveled. When it becomes necessary to reseat the valves, it will be found that it is much easier to take the nicks out of a small beveled seat than a large flat one.

I prefer to use beveled seats if possible, but under some conditions this cannot be done—for example, where it is necessary to use thin flat disks as in air pumps—but for boiler-feed pumps I believe from experience that metal disks with beveled seats are the best.

Concord, N. H. H. C. WILLIAMS.

Whether a valve that has a beveled or one that has a flat seat is better depends a great deal on what service the pump is used for. I believe that the beveled seat is better for boiler-feed pumps or others handling hot water at high pressure, but there may be no advantage when used for cold water and especially for low pressure.

Flat valves, 3 or 4 in. diameter, of brass or hard rubber have a tendency to spring when heated, and considerable leakage may occur even though the seat is in good condition. Small obstructions are more apt to lodge on a flat seat in making the two right-angle turns necessary to get through, especially the discharge valves. The beveled seat has the advantage that it is very narrow—usually not over ⅛ to ¼ in.—and as the flow is more nearly straight, there is less tendency for any obstruction to stick on the seat. The lift of the bevel valve is usually less than that of the flat one, smaller valves may be used for a given capacity, and the beveled seat is easily ground in to a tight fit.

The foregoing is the practical and not the theoretical side of the question. J. C. HAWKINS.

Hyattsville, Md.

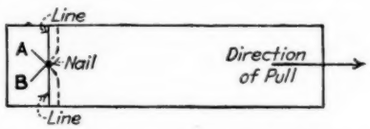
Efficiency of Unsymmetrical Riveted Joints

Replying to A. A. Adler's criticism of my article on "Efficiency of Unsymmetrical Riveted Joints" in the Aug. 8 issue I may say that the example given of the distribution of the stress over the rivets in a double-riveted butt joint was intended merely to show that all the factors affecting the efficiency are not taken into account in the usual formulas for calculating the efficiency of riveted joints and that the results obtained by such formulas are therefore not absolutely correct. The actual distribution of the stress over the rivets, as Mr. Adler points out, is materially affected by the deformation of the plate, and also, I may add, by bending of the rivets, but if these factors had been taken into consideration the calculation would have become very complex. To keep the article within a reasonable length it was necessary to omit, or at least to simplify, all calculations as far as possible. I do not think, however, that in practice the stress is distributed anywhere nearly as uniformly as Mr. Adler's letter would appear to imply. It is well known that there is a much greater tendency to leakage at rivets in the outer rows of wide joints than at rivets in the inner rows, and the outer rivet holes will frequently be found to be elongated after the boiler has been in use, thus showing that there is a greater stress on the outer rivets than on the inner rows.

Referring to the objections to the conclusions deduced from an elastic rubber model, I may point out that

Hooke's law, "The deformation produced is proportionate to the stress, and vice versa," holds equally good within the elastic limit for all materials, and that within this limit the distribution of stress is not affected by the tensile or shearing strengths of the material. The latter affect only the ultimate strength of the joint, and unless the stress on the joint is made so great as to entirely alter the stress conditions by materially distorting its shape, it matters not, so far as the distribution of the stress is concerned, whether the material is rubber or steel.

In the illustration given in Fig. 2 of the article, and herewith reproduced, the only reason why the stress appears more concentrated near the nail is because of its small diameter as compared with a rivet in a flat brace. This is in strict accordance with the formula



RUBBER-BAND EXPERIMENT

given for computing the stress at the sides of a rivet hole. If the experiment is made with a piece of rubber, say ¼ in. thick and a peg ¾ in. in diameter, the distribution of the stress due to a pull on the rubber strip sufficiently great only to produce a slight stretching of the rubber will be the same as in the case of a flat steel brace of the same thickness secured by a rivet of equal diameter.

The object in writing the article was to endeavor to show that unsymmetrical joints are inferior to symmetrical joints on account of the bending stresses produced in them by the tension in the boiler shell and that the efficiency of such joints, especially when there are three or more rows of rivets, is usually overrated.

I am pleased to note that Mr. Adler agrees with me that it is unfortunate that these joints are illustrated in the A.S.M.E. Code to the almost complete exclusion of symmetrical joints, thus creating the impression that they represent the best type of construction.

In reply to Mr. Adler's request, I regret that an investigation of the law of variation of stress in the plate and the method of deducing the formula given in the article for calculating the stress at the sides of a rivet hole would be too lengthy to include in this letter, but if the editor deems the matter of sufficient interest to afford the space, I shall be glad to write another article dealing with this subject. R. N. BLACKBURN.

Regina, Sask., Canada.

Viscosity of Cylinder Oils

Referring to the opening paragraph of Mr. Whitlow's letter on page 598 of the Oct. 24 issue, permit me to disclaim having made the assertion he attributes to me. What I said was that the contention that viscosity is of no value as a criterion of lubricating quality has been made *chiefly* (not "only") by persons interested in the exploitation of cylinder lubricants containing animal fats. This contention has been put forward by a few others, but relatively few.

Mr. Whitlow strikes the keynote of the whole question in the fifth paragraph of his letter: "The general result of investigations . . . is to discredit the value usually attached to the characteristics of oils *in the manner in which they are conventionally handled and stated.*" The italics are mine. Viscosity, or any other characteristic, considered alone is practically worthless as a criterion; and the viscosity of cylinder oils at ordinary temperatures is a rough and unsatisfactory guide. With the percentage of animal or vegetable fat limited to a low quantity, however, and the viscosity taken at a high temperature, this characteristic is a most valuable criterion when considered in combination with others.

The specifications under which the City of Atlanta buys all classes of lubricating oil prohibit all acidity, any filler or thickener, and more than 3 per cent. residue or 5 per cent. evaporation. The flash point, fire point and cold point are limited according to the class of oil. No restriction is specified for viscosity, gravity or saponification, but the oils are tested for these characteristics, as well as the other seven mentioned, and judgment of the relative quality is based on the combination—not any one point.

All lubricating oils except cylinder oils are tested for friction coefficient on a Thurston machine, and this is considered the principal factor in quality. Cylinder oils are not tested on the Thurston machine, because the cylinder temperature and pressure conditions cannot be even approximated. Viscosity tests of all oils are made at the temperatures at which they are used, as nearly as those temperatures can be determined. In testing at temperatures above 210 deg. F., the jacket of the viscosimeter is filled with glycerin instead of water, and a special nozzle, of smaller orifice than those for lower temperatures, is used.

As stated in my original communication on this subject, we have found that, under the conditions here outlined and within certain ranges, differing according to the kind of oil (whether low-pressure or high-pressure cylinder, heavy or light engine, etc.), the ratio $\frac{\sqrt{\text{viscosity}}}{\text{specific gravity}}$ is an excellent criterion of the lubricating value.

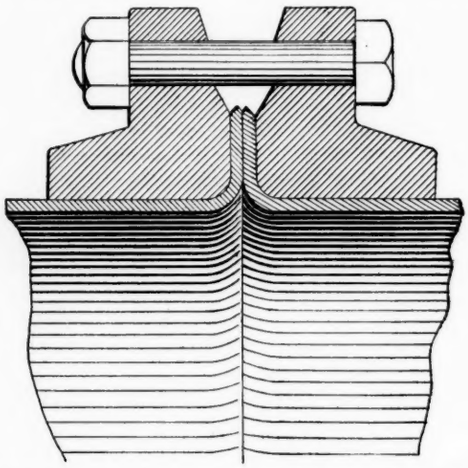
Atlanta, Ga. CECIL P. POOLE,
City Mechanical Engineer.

Welded Steam-Pipe Joints

The higher steam pressures coming into vogue give rise to new methods of joining the steam piping. One of more than passing interest is being tried out by the Commonwealth Edison Co. at two of their stations. They have discarded the use of gaskets altogether and weld the flanged ends of the pipes together. The joints are of the Van Stone type, but have somewhat heavier flanges, the face of which is beveled off as shown, so that access may be had to the flanged part of the pipe for welding. The inner faces of the outer edges of the pipes, between the two flanges, are also beveled off to form a V-shaped groove, when the pipe is in position. This groove is filled up with soft Russian iron by means of an oxyacetylene torch.

The weld is to prevent steam blowing out and plays no part in the mechanical strength of the joint; the flanges are sufficient for that purpose. These joints were first tried on the 18-in. main steam header at Fisk Street carrying steam at 200 lb. pressure and 150 deg.

F. superheat, and proved so successful, in that it has given no trouble of any kind to date, that it was decided to use it for the complete steam-piping system in sizes 7-, 8-, 10-, 12-, 15-, 17- and 19-in. for this company's Northwest Station. The main header is about 200



VAN STONE JOINT WELDED AT OUTER EDGE

ft. in length and carries steam at 250 lb. and 225 deg. superheat. Since turbines are already being built for 350 lb. pressure and at least one manufacturer stands ready to guarantee turbines operating at 500 lb., it would appear that the use of such welded joints will find increasing use in the stations of the future.

Chicago, Ill. R. K. Long.

The Shunt Was Shunted

When a continuous-current compound-wound generator supplies a load that is characterized by sudden and violent fluctuations, there is a tendency for the generator to spark during the period of fluctuation. This is because compound-wound machines generally are liberally over-compounded, in which case it is necessary to shunt the series-field winding in order to reduce the compounding to the desired value. For steady loads or for loads of which all variations take place gradually, noninductive shunts of the usual type are satisfactory, for there is no current-selective action that may be influenced by the relative self-induction of the series-field winding and of its shunt. Under the condition of a sudden load change—an increase for example—the self-induction of the series winding causes the greater part of the current increase to go through the noninductive shunt instead of through the series field winding. This momentarily weakens the field magnetization relative to that of the armature, thereby so distorting the commutating field as to produce poor commutation. In order to overcome this tendency, a shunt that has considerable self-induction is connected in series with the noninductive shunt. The combined series resistance of the two is equal to the shunt resistance required to give the desired degree of compounding at full steady load. The self-induction of the inductive shunt is such as to maintain the proportional division of current between the series field and its shunt during periods of load fluctuation.

A generator that was to supply energy for a number of mill motors, among which were several large ones, was installed by the owner, and soon after being placed in operation, it not only developed bad sparking whenever one of the large motors was started, but it failed to maintain the voltage for which, presumably, it had been compounded. An inspector investigated and found that the trouble was due to the following conditions: In trying the brush-shifting device, someone had failed to return the brushes to their original position; this had slightly lowered the voltage on steady loads, and with the idea of increasing the voltage, an improvised german-silver shunt had been added to the existing shunt. Instead of connecting it in series with the existing shunt, however, it was connected across the inductive shunt, practically short-circuiting it inductively, so that proper division of current impulses between the series field and its shunt as a whole ceased to be obtained.

By simply removing the improvised shunt and shifting the brushes to the marked position, the original compounding on steady loads was restored and sparking during fluctuations of load was eliminated.

Brooklyn, N. Y. E. C. Parham.

Globe Valves Opening Against Pressure

In a certain plant a number of 6- and 8-in. globe valves were used where the pressure was sometimes under and sometimes over the disk, and occasionally they would pull off the stems so the valves could not be opened. Some of these valves were repaired by upsetting or riveting the end of the stem, but later repair was made by threading the end of the stem a short distance with a pipe die and screwing on a ring sawed from a pipe coupling and riveting it securely. It is best to use a left-hand die and coupling.

An 8-in. globe valve opening against 120-lb. pressure requires a pull of over three tons to lift the disk from the seat, therefore it is apparent that globe valves should not open against the pressure unless a bypass is provided. G. E. Miles.

Green River, Wyo.

Removing Ruptured Boiler Tube

It is general practice, when a tube ruptures in a water-tube boiler, to cut out a sufficient number of tubes to enable workmen to get to the one that is defective, as it is usually bagged so that it is not possible to pull it through the header. This method is not only expensive, but requires a great deal of time, keeping the boiler out of service.

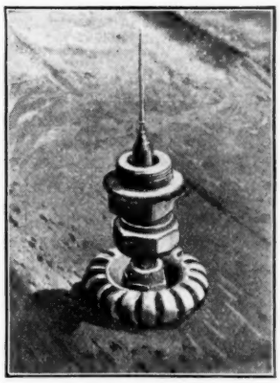
A few weeks ago a tube ruptured in the eighth row from the bottom and near the center header in a B. & W. boiler. To cut this tube out it would have been necessary to take out fifteen more, and the boiler could not be kept out of service so long, so it was decided to pull the next tube, then use an oxyacetylene welding outfit to cut the bagged one in small enough pieces to be taken out through the header. The whole operation—taking out the two tubes and replacing them with new ones—required but 12 hours for two men, saving a lot of time and expense.

Louisville, Ky. Joseph L. Brown.

Double-Ply Leather Belts should not be used on pulleys much less than 12 in. diameter and three-ply on pulleys less than 20 in.

Keeping Lubricator Feed Nozzles Open

The oil-feed nozzle of lubricators sometimes gets stopped up, necessitating the removal of the regulating-valve bonnet and clearing it with a wire. The illustration shows a piece of slender steel wire fastened to the end of the valve to extend up through the nozzle nearly to the tip. Each time the valve is used, the wire loosens any small particles of foreign matter that might adhere to the walls of the tube and stop the passage. The wire should not be large enough to interfere with the flow of the oil. This kink is similar to the extension found on some glass gage valves intended to keep the passages clear of scale, and is well worth the time required to fix.



EXTENSION ON VALVE STEM

F. W. Bentley, Jr.

Missouri Valley, Iowa.

Laying Out Quarter-Turn Belts

There is generally a great lack of positive information in regard to the process of laying out quarter-turn belts. Fig. 1 represents a unique paper model, where *D* is the driving and *C* the driven pulley, turning in the direction of the arrows as shown in Fig. 2. By laying out this model on Bristol board or thick drawing paper to any scale and then folding upon the line *A* one may readily get the center lines so that the belt will run satisfactorily from one pulley to the other, in proper alignment at all angles.

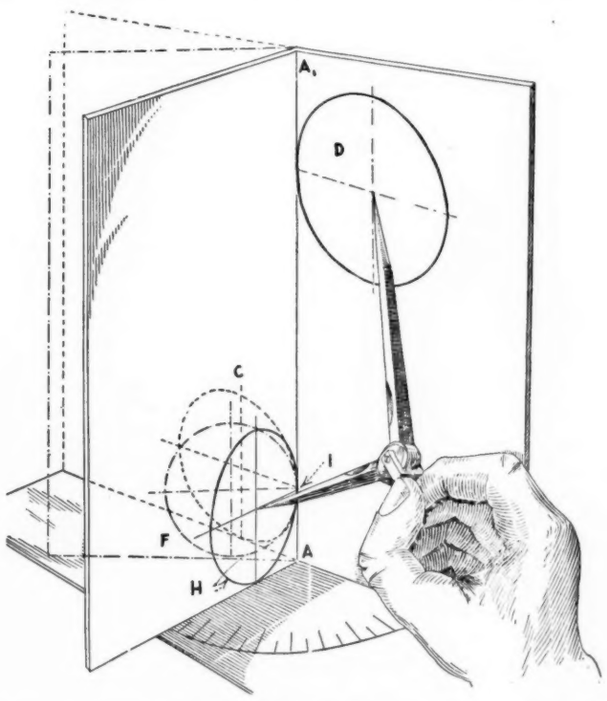


FIG. 1. MINIATURE PAPER MODEL OF QUARTER-TURN LAYOUT

It will also be discovered that the driven pulley may be placed at any angle to the driving pulley, as shown in Fig. 1, where the plans of the two pulleys are represented. The driving pulley *D* is shown over the driven pulley *C*, which is first in line with the driving pulley, then shifted around to a right angle *H* with the driving pulley, always being careful to keep the line *A* and A_1 of the two pulleys coincident with the lower at *I*—considering the line *A* and A_1 to be vertical, to get the idea clearly, after which it may be varied from the original to suit any special conditions that may develop. By a little experimenting with the model, one may soon become expert in laying out drives to run at any desired angle.

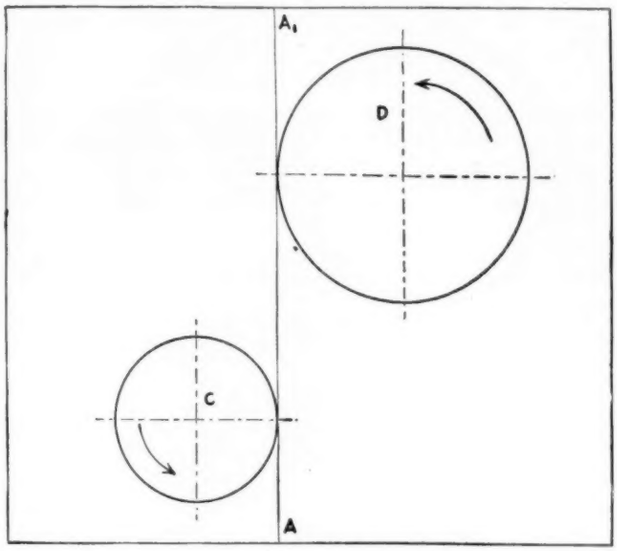


FIG. 2. STARTING TO LAY OUT A DRIVE AT ANY ANGLE

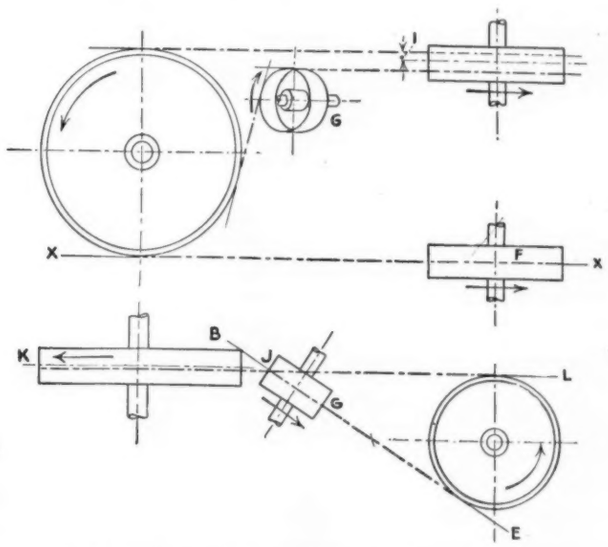


FIG. 3. GUIDE PULLEY USED IN SPECIAL CASES

Where the driving or driven pulley is very large and the other pulley is small, as in Fig. 3, and where the distances between the shafts are quite short, it becomes necessary to introduce an idler or guide pulley *G*. It is then advisable to lower the driven pulley, as shown

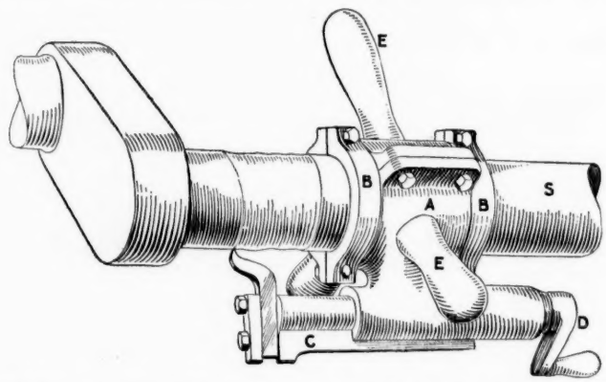
at *I*, and have the distance between the top of the idler and the tension side of the belt not more than three inches, the closer the better without touching.

To locate the position or idler *G*, a line from *B* to *E* represents the center line of the idler-pulley face. It will be observed that where the lines *BE* and *KL* intersect at *J* is the location for the edge of idler pulley *G*. Its shaft, of course, must be at right angles to the line *BE*. The idler is not adjustable to take up the slack of the belt, but must be fixed permanently. Where the distance between the pulleys will permit and the belt is not so wide as to cause it to be overstrained, the idler *G* may be dispensed with and the driven pulley lowered to position *F*, with the center line of its face coincident with that of the line *XX*.

Hand Turning Tool for Shafts

The main engine journals of a tugboat that I was running were too low and had to be rebabbitted, so we laid up for repairs in a small town on the Connecticut coast and started to raise the boxes.

We were surprised at the condition of the shaft, which was cut so much that it required turning. The foreman of the shop said he could not take care of the job any more than a watchmaker could turn up a 6-ft. propeller,



HAND-OPERATED TOOL FOR TRUING UP ROUGH SHAFT

as his largest lathe was only 6-ft. centers. Before leaving the boat, however, he told me that he would consider the matter overnight and see what he could do. When he came the following day, he measured the diameter of the shaft, the length of the bearing portion and returned to the shop with a rough sketch. Patterns and castings were made, and two days later he returned to the boat with the device shown in the illustration.

A split casting *A* bolted around the good portion of the shaft *S* so that it could revolve was held from sliding endwise by two split collars *B* clamped on the shaft. A tool-supporting slide *C* dovetailed in the casting *A* was moved ahead by the screw and handle *D*. Three handles *E* were screwed on the casting by which the tool was rotated with the shaft suspended on the tackles. The tool was set at the beginning of the scored portion and the rig turned by means of the three handles. The tool was advanced by the screw *D* at every revolution.

The job was well done and at a very reasonable price compared to the time and cost to remove the shaft from the boat to a large shop. The device became part of our equipment and has been used on other jobs.

Chester Park, L. I. A. J. Saeul.

Haste Often Makes Waste

The electric-lighting plant in this city is part of an interconnected system. Current is distributed for lighting service in the usual way, and motor generator sets furnish 250-volt direct current for a factory load 24 hours per day, and 1,200-volt direct current for an electric railway. One small and two large water-tube boilers are installed.

Recently one of the large boilers was shut down for cleaning but before the work was done, the arch in the firebox of the other one dropped and the chain grates of the stoker had to be pulled out to make repairs. Work on the first boiler was rushed, and it was fired up late in the day.

About 8 p.m. I chanced to be passing and noticed that only the motor-generator sets were running and found the night engineer back of the boiler waiting for it to drain to replace two handhole gaskets near the bottom that were leaking badly, owing to the hurried work getting the boiler capped up. Fortunately the other plants of the system, although carrying snug loads, were able to handle the burden without trouble until the boiler was again put in service. But the time of the crew in the idle plant was a loss, to say nothing of the possibility of trouble from the overload on other parts of the system.

The incident showed the advantage of care, even in hurried work, and incidentally of an interconnected system; for a city of 7,000 people was lighted, a large factory furnished with power and light and an electric railway operated with current generated fifty miles away.

Charles City, Iowa. C. V. Hull.

Government Regulation of Power-Plant Practice

I understand that in France and Germany before the war no building could be erected without the height and area of the chimney or flue first being passed upon by a government official. This precaution was taken in order to stop the waste of the country's fuel, which each year becomes more expensive and more difficult to get. The reason, of course, back of this law is that every chimney should be suited to the furnace, boiler, etc., for which it is built, so that it will furnish somewhere near the correct amount of air. If burning coal does not get sufficient air, it simply decomposes and gives off only a fraction of its heat; likewise, too much air or draft causes a large percentage of heat to pass up the chimney and, if excessive, would chill or put the fire out.

While coal is cheaper and much more plentiful in this country, we should have a law governing the operation of steam plants, which would prohibit passing the waste gases into the chimney at more than 500 deg. F.; that is to say, making it imperative that the boiler plant be properly designed, equipped and operated. Flue gas at 500 deg. contains approximately 20 per cent. of all the heat originally in the coal—an enormous waste. While it is true that there is no way of recovering all this waste heat, a good percentage can be profitably saved, and the plant should not be operated at such ratings as to cause avoidable waste.

I would like to have this matter discussed and, if it has ever been brought up before in *Power*, to know when.

New York City. F. J. McMahon.

Inquiries of General Interest

Height of Pumping Water—How high can water be raised with a direct-acting steam pump having a steam cylinder 8 in. diameter and a water cylinder 16 in. diameter if operated by steam at a pressure of 90 lb. per sq.in. and the suction water is at a pressure of 40 lb. per sq.in.? P. M.

Neglecting the reduction of the piston areas due to the presence of the piston rod, the areas of the steam and water pistons would be to each other as the squares of their diameters; that is, as 8^2 to 16^2 or as 1 to 4. Neglecting friction, a steam pressure of 90 lb. per sq.in. on the steam piston would exert $90 \div 4 = 22.5$ lb. per sq.in. on the water piston; and as the water piston would also be urged by a suction pressure of 40 lb. per sq.in., the total water pressure obtained in the water cylinder would be $22.5 + 40 = 62.5$ lb. per sq.in., which without frictional resistance would force the water on the discharge side of the piston to a height of $62.5 \times 2.3 = 143.75$ ft. The height to which the water actually could be pumped would depend on the percentage of steam pressure lost in overcoming friction of the pump and the percentage of loss of water-cylinder pressure required for overcoming the friction of the water passages and of the discharge pipe.

Change of Frequency on Induction Motors—What would be the effect of operating a 25-cycle induction motor on a 60-cycle circuit or vice versa? M. A. L.

In general, the normal capacity of a 25-cycle induction motor operating on a 60-cycle circuit will be approximately the same. The starting torque will be materially reduced, also the pull-out point. The speed will be increased in a ratio of 25 to 60; that is, a two-pole motor operating on a 25-cycle circuit will run 1,500 r.p.m., while on a 60-cycle circuit it will run 3,600 r.p.m. Such an increase in speed might overstress the rotor. A 60-cycle induction motor operating on a 25-cycle will require theoretically 2.4 times the magnetizing current, but this will depend on the design of the motor, and it is probable that the exciting current would be considerably above this figure. The effects are excessive heating of the machine, a reduction of capacity, a low power factor and the speed will be reduced in a proportion of 60 : 25.

Although 25-cycle motors have in some cases been operated on 60-cycle circuits with considerable success, it is doubtful if a 60-cycle machine will operate satisfactorily on a 25-cycle circuit. In general, either practice is not to be recommended.

Transmission of Heat Through Iron Plates—What is the rate of transmission of heat through iron plates ¼ in., ⅜ in. and ½ in. thick from water at a temperature 212 deg. F. on one side to water at a lower temperature on the other side? C. W. B.

The rate of transmission of heat through a metallic plate having water on each side of the plate at different temperatures depends on the condition of the metallic surfaces and the activity of circulation of the water on both sides of the plate. According to experiments by M. Peclet, when the fluids in contact with the surfaces of the plate are not changed by artificial means, the rate of conduction is not only the same for different metals, but also for different thicknesses of the same metal. This uniformity is ascribed to the presence of stagnant films of water adhering to the surfaces of the plate which neutralize its conductivity. When the surfaces of the metal are perfectly clean and the water in contact thoroughly and constantly changed, M. Peclet found the rate of transmission from water to water through a plate or bed of cast iron 1 in. thick was 225 B.t.u. per hr. per sq.ft. for 1 deg. F. difference of temperature; and as the conductivity is inversely as the thickness, under the conditions stated the rate of transmission for ¼-in. plate would be 900 B.t.u., for ⅜-in. plate 600 B.t.u. and for ½-in. plate 450 B.t.u. per hr. per sq.ft. per deg. difference of temperature of the water.

Quantity of Air Required To Burn a Ton of Coal—What quantity of air would be required to burn a ton of coal consisting of 69.8 per cent. carbon, 5.26 per cent. hydrogen and 8.35 per cent. oxygen? J. W.

The theoretical weight of air required for combustion is approximately determined by the formula,

$$W = 12C + 35\left(H - \frac{O}{8}\right)$$

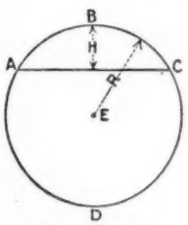
in which W is the weight, in pounds, of air required per pound of the coal, and C, H and O represent the parts of a pound of carbon, hydrogen and oxygen respectively in a pound of the coal. By this formula a pound of coal of the given analysis would require $(12 \times 0.698) + 35\left(0.0526 - \frac{0.0835}{8}\right) = 9.85$ lb. of air, or a ton of 2,000 lb. of the coal would require, theoretically, $9.85 \times 2{,}000 = 19{,}700$ lb. of air. For practical purposes it is unnecessary to calculate accurately the amount of air required to burn coal, and as most fuels require theoretically between 11 and 12 lb. of air per lb. of coal, it is usual to consider that each pound of coal burned will require 12 lb. of air. But to insure more perfect combustion by supplying an abundance of oxygen, it is necessary to admit more air into the furnace than is theoretically required to burn the coal. It therefore is usual to supply twice as much air, or about 24 lb. per pound of coal, with natural draft, and one and one-half times, or about 18 lb., when forced draft is used. One pound of air at 62 deg. F. has a volume of 13.14 cu.ft., and for burning a ton of coal with natural draft the quantity of air actually required is about $2{,}000 \times 24 = 48{,}000$ lb., or $48{,}000 \times 13.14 = 630{,}720$ cu.ft., and about three-quarters as much when forced draft is used.

Content of Cylindrical Tanks—When a cylindrical oil tank with bumped heads rests horizontally on its side, how can the capacities for various depths be determined? G. A. C.

For practical purposes computations of the capacities for different depths will be within the errors of measurements if it is assumed that the cylindrical portion of the tank has its length increased by one-half the bumped height of both heads and that the bumped heads are replaced by flat heads. The volume contained at any depth then will be the cross-sectional area of the flooded section multiplied by the assumed length of the tank. For depths less than the radius of the cylindrical portion of the tank, the cross-section of the flooded portion will be a segment of a circle whose area is given by the expression,

$$\frac{4H^2}{3}\sqrt{\frac{2R}{H} - 0.608}$$

where H = the depth and R = the radius of the tank. When the depth equals the radius, the cross-section will be a semicircle, or ½ of $(R^2 \times 3.1416)$. For depths greater than the radius of the tank, the cross-sectional area of the unflooded portion is to be deducted from the cross-sectional area of the tank. In the figure ABCD represents the cross-section of the tank, ACD the flooded portion and ABC the unflooded portion. Then, as before, R = the radius, but H = the height of the unflooded portion, or the diameter of the tank minus the depth of the flooded portion. Therefore, for a cross-section whose depth is greater than R and where H = the height of the unflooded portion, the area of the flooded portion is given by the expression,



COMPUTING AREA OF SEGMENT

$$(R^2 \times 3.1416) - \frac{4H^2}{3}\sqrt{\frac{2R}{H} - 0.608}$$

For computing a large number of cross-sectional areas, as for laying off a rod for gaging the capacity of a tank, use may be made of mathematical tables that give areas of segments of circles for different values of H when the diameter = 1. To use such a table, divide the actual value of H by the diameter of the tank and multiply the tabular area that corresponds with this quotient by the square of the actual diameter. When the dimensions are all taken in inches, the area of the assumed section will be given in square inches and this multiplied by the assumed length of the tank in inches will give the volume for the assumed depth, in cubic inches. To reduce the result to American gallons, divide the volume in cubic inches by 231 and for British gallons divide by 277.274.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—Editor.]

Chimney and Mechanical Draft*

Boiler and furnace practice has changed to such an extent within the last few years that the generally accepted ideas of a few years ago are now largely obsolete. Most large and many small plants are now equipped with mechanical stokers of some one of the numerous types, particularly in those centers where room is limited and smoke abatement laws are enforced. While chimneys alone are still largely depended upon to produce the draft necessary for the combustion of fuel, the tendency is strongly toward supplementing them with mechanical equipment. The effective draft, or that which is used for getting the air through the fuel bed, may be produced in three general ways: First, by the natural draft of the simple chimney; second, by induced draft; third, by forced draft. But when the whole of the boiler installation is considered, several combinations of these may be desirable or necessary.

Where natural draft alone is depended upon, it must be of sufficient intensity to overcome all the resistances from the top of the stack to the ashpit door. With a given height of stack the available draft is practically fixed, although it varies some with the temperature, state of the weather, barometer, etc. A chimney of moderate height cannot supply a sufficient draft to maintain what is now looked upon as a high rate of combustion. For conditions under its maximum, natural draft is easily controllable by means of dampers so that the furnace conditions can be maintained at the most economical point. The principal recommendations of a chimney are its simplicity of operation, its need for comparatively little care and attention and its being always on the job. Its thermal efficiency is low, but the other advantages offset that in cases where high draft intensity is not needed.

To attain a strong draft by means of a chimney is prohibitive because of the great height involved and the cost. It is therefore necessary to turn to other arrangements, such as an induced-draft system, wherein the suction is created by a fan. The nature of this draft is exactly the same as chimney draft, but it has the advantage of being susceptible to accurate control so that the furnace needs may be readily met. It makes possible the installation of economizers to lower the temperature of the escaping gases and effect a saving in fuel.

At the present time a combination of forced draft and either natural or induced draft is being used more and more. Many of the later stokers are operative only with forced draft and are designed for high rates of combustion. Forced draft alone is not a good arrangement for the reason that it creates a plenum throughout practically the whole installation and makes leaks outwardly through the interstices in the setting likely. By providing a stack of proper height or an induced-draft fan, the necessary suction draft may be supplied to overcome the resistance of the gas passages, leaving the forced-draft fan to supply the pressure necessary to overcome the resistance of the fuel bed. This produces that desirable condition, balanced draft, the pressure above the fuel bed being practically zero or preferably a little on the minus side; that is, the draft gage should show a slight vacuum.

The amount of fuel burned under a boiler largely determines the amount of steam that will be generated. There is, of course, a more or less well-defined relation between grate area and heating surface, although this varies between rather wide limits. The heating surface of boilers in many instances is being worked far harder than was supposed possible a few years ago. This has been brought about by the introduction of furnaces capable of high rates of combustion, with the result that the temperatures of the escaping gases are too high for the proper economy, rendering necessary the use of feed-water economizers of proper capacity. Such installations are economical as to heat absorption, in interest charges resulting from a smaller investment and in floor space. The great objection to these types of installations is the complication due to more operating parts, from which the plain chimney of our forefathers is entirely free.

The need for a high intensity of suction draft in connection with forced draft, when high capacities are used, is not generally recognized. When it is realized that the draft required to cause the flow of gas increases as the square of the velocity, it will be seen that this is important. If one-half inch in the uptake is sufficient for satisfactory operation at one capacity, doubling the capacity at least doubles the quantity of gas, which means that the gases must flow twice as fast, assuming that the furnace efficiencies and temperatures are the same in both cases. Theoretically, therefore, it would take four times as much draft, or two inches as measured by the water column. It is on this rock that many engineers have come to grief in that the attainment of their desire to get a high capacity out of their boilers has not been possible because after reaching a certain rate of combustion there is a plenum created in the furnace and through the setting, resulting in furnace gases coming out through the numerous interstices that inevitably exist, owing to the insufficiency of the available suction draft to pull the gases through the boiler and breeching.

The balanced condition is highly advantageous in another way. The greater the difference in pressure between the inside of the setting and the outside, the greater the amount of air drawn in through any cracks or openings that exist, as well as through the porous setting. The less these differences in pressure the less air will be drawn through. All such air is only detrimental and can do no possible good. Every effort should be made to prevent these leakages if the boiler plant is to be run at its highest efficiency.

It is now common to inclose the entire setting in a sheet-iron casing, carefully fitted and made as tight as practicable. Special preparations for coating the outside of brick walls and stopping up cracks are on the market, the goal of all being the same—the exclusion of unnecessary and objectionable air from the interior of the setting.

To attempt to force a grate to burn more fuel than the draft available makes possible results inevitably in waste and inefficiency through the formation of carbon monoxide instead of carbondioxide, so that where draft cannot be increased beyond a certain point, care must be used not to try to force a higher rate of combustion than can be efficiently maintained.

Every set of boiler and furnace conditions has its most economical point of operation, which cannot be exceeded with satisfactory results, but this is true within much narrower limits of simple chimney draft than with either forced or induced draft with the accompanying possibility of widely varying adjustments.

The tendency of modern boiler practice is decidedly toward setting boilers well above the grates so as to give a large combustion space. It is pretty generally recognized that the heat of combustion should all be freed before any of the gases come in contact with the heating surface of the boilers, which in relation to the gases is the cooling surface. A large, high furnace and combustion chamber is essential to attain this end, in addition to which the resistance to flow is minimized.

It is important that the breeching should be as short and direct as possible. Square turns of any sort should be avoided, as all such increase the resistance. The well-known principles governing the flow of water apply almost as universally to the flow of gases. Easy curves should be provided and all corners and dead spaces avoided. Areas must be properly proportioned to the volume of gases to be cared for, and this is particularly true in those plants where high overload capacities are contemplated.

Modern boiler practice seems to indicate that for the larger and more important plants at least, something more than ordinary chimney draft is needed. Where simplicity of installation is paramount, simple chimney draft cannot be improved upon. As applying to the average plant throughout the country, use every means to prevent air from entering the gas passages other than through the fuel bed; see that the loss of draft due to restricted, improperly designed and obstructed flues is reduced to a minimum, so that the boiler and furnace will be in condition to give the maximum efficiency.

*Abstract of paper read by E. R. Fish, of the Heine Safety Boiler Co., before the eleventh annual convention of the Smoke Prevention Association, St. Louis, Mo.

Fatal Blowoff-Tank Explosion

A cast-iron blowoff tank exploded at the Toledo, Ohio, city water-works pumping station on Friday afternoon, Oct. 27, causing the death of William Grassman.

The tank was 48 in. high by 30 in. diameter and was probably originally ¾ in. thick, but it had been in service more than 20 years and had corroded on the exterior side so that at the thinnest place at the break it was not more than $\frac{5}{16}$ in. It was set half below the floor line, and it was in this lower half that the corrosion and failure occurred, the break extending all the way around near the bottom. The upper part was thrown upward about 20 ft. after tearing loose from the 6-in. sewer pipe connection and two 2½-in. blowoff connections. It is not stated, in the press or other reports at hand, whether the tank had an atmospheric relief pipe or not. The pressure on the boiler being blown down was 150 lb., but it is not likely that the tank had anything like that pressure in it unless the outlet was stopped, as the sewer outlet was nominally 6 in.

The victim had been employed at the station as fireman, oiler, etc., for eight or 10 years, and there is therefore no suggestion of a mistake in manipulation.

Power Machinery Used in South Russia Coal District

According to the October, 1916, issue of "Russia," a summary of steam machinery used in the South Russia coal district—the Donetz Basin—shows that the number of steam engines during the five-year period from 1909 to 1914, doubled, increasing from 1,235 to 2,486; the total horsepower also doubled, rising from 128,114 hp. to 254,221.

The state of the electrical equipment in the district is indicated by the following figures on electrical generators, representing replies from 53 different firms in 1909 and 51 firms in 1914. In 1909 there were in operation 159 generators having a total capacity of 22,889 kw.; in 1914 the number decreased to 148, but the total capacity increased to 54,551 kw., showing that a large percentage of the machines that were in use in 1909 were discarded during the following five years and were replaced by larger and more efficient equipment. Alternating-current equipment, principally of the three-phase type, has recently been substituted by many firms for the earlier direct-current generators. With this change the transmission voltage has, of course, been much increased.

Generating electric energy by alternators and stepping up the voltage by means of transformers for transmitting over considerable distance has been quickly recognized as very advantageous in this field and has already created a large demand for this type of equipment in the Russian market.

Out of 159 generators in use by the firms in 1909, there were 60 alternators, or 38 per cent.; out of 148 generators in use in 1914, 80, or 54 per cent., were alternators.

The use of current of higher voltages in comparison with the equipment of a few years ago is marked in the returns for 1914; in that year out of a total of 80 alternating-current generators only 8 were of lower power than 500 kw., while 45 alternators (over half the total number) were of more than 2,500 kw. Two alternators were of 5,500-kw. capacity.

Wide substitution of the modern incandescent lamp for the old-style carbon bulb, as well as for arc lights, is one of the most interesting features. In 1909 there were 24,830 incandescent lamps in use. This had increased to 39,320 in 1914. The number of arc lamps in use in 1909 numbered 914, which number had decreased to 520 in 1914.

Another significant point in this summary is the rise in the total power of electric motors from 19,081 kw. in 1909 to 46,238 kw. in 1914. The third point is the rise in the average power for each firm from 360 kw. in 1909 to 963 kw. in 1914. The number of motors used by the 53 firms giving information was 793 in 1909; this increased to 1,298 for 48 firms giving information in 1914.

These general figures for the Donetz region represent the situation in the other coal districts of Russia. Russian mining men began to realize years ago the usefulness of electrical equipment of various kinds in mine work, and the success of their first trials with it led to a steady expansion of its application. It is significant, from the point of view of the foreign trade of the United States, that German electrical equipment was used in the Donetz Basin when the use of electricity began to expand; and the later increase of equipment was supplied almost wholly by imports of generators and other electrical materials from Germany. This importation from Germany was, of course, stopped by the war. After the war there will be a greatly increased demand—as indeed there is now—for additional electrical equipment. There is an important opportunity here for the American manufacturer.

Recent Court Decisions

Risk Assumed by Engineer—The Pennsylvania Supreme Court recently denied the right of an engineer in an artificial ice plant to recover damages from his employer for injuries sustained through his foot slipping and being crushed by the revolving crankshaft of his engine while he was filling one of the oil cups. The ground of the decision is that the engineer knew better than anyone else the risk of oiling the engine in the position he assumed, and that therefore he accepted the risk. In reaching this conclusion, the court applies the following general principles of law:

Where risks are incidental to employment and, from his familiarity with them, an employee is equally as familiar with them as his employer, or more so, no liability attaches to the employer for injuries sustained by the employee in consequence of his exposure to such perils. An employee who continues at work in a place that he knows to be dangerous, taking his chances against injury, must bear the consequences of his own rashness. "The machinery being under plaintiff's control, it was his duty under the circumstances to have stopped the engine before attempting to oil it, as he knew it was dangerous to do so while it was in motion, and this rule applies although the machine was not guarded." (Card vs. Stowers Pork Packing and Provision Co., 98 "Atlantic Reporter," 728.)

Workmen's Compensation Act Applied—A stationary engineer was fatally stabbed by an Italian laborer when the former apparently attempted to eject the latter from a boiler room on his calling the engineer a vile name. It appears that the ill feeling arose through the engineer having reported a breach of duty on the Italian's part to their common employer. The deceased engineer's widow applied for an award under the New York Workmen's Compensation Act on the theory that the fatal injury was received in the course of her husband's employment, but the State Industrial Commission has disallowed the claim, saying:

There can be no doubt that an injury growing out of a personal assault may be an accident within the meaning of the compensation law under certain circumstances. It is, however, difficult in many cases to determine whether such an accident arose out of and in the course of the employment. . . . I am unable to see how the injury to Mr. Slane arose out of the employment. It is true that the injury was received on the plant of the employer and during the time when Mr. Slane was employed, and was at the hands of another employee. It is also true that the opprobrious epithet which the Italian applied to Mr. Slane would warrant in the eyes of most men chastisement at the hands of the man to whom it was applied. So far as appears it was this natural impulse to resent such an epithet which caused the deceased to enter into the fracas with the man who used the term. I think the fair inference from all the facts is that Mr. Slane in a moment of anger attempted to forcibly throw the Italian out of the engine room and for that purpose was the aggressor in the assault. There was certainly nothing in Mr. Slane's employment which would cause him in the interest of the employer to resent the epithet hurled at him to the point of committing an assault, and while he would probably be excused by most men in yielding to the natural impulse to chastise his opponent, I think that in so doing he stepped out of his regular employment and that the claim for compensation must be disallowed on the ground that the accident did not arise out of the employment of the deceased.

Boiler Legislation Meeting

Ohio, through its Industrial Commission and under the seal of the state, has issued a call for a meeting of all states of the Union to discuss boiler legislation, to take place the week of Dec. 4 at the Hotel Willard at Washington, D. C. The report of Chairman Thomas E. Durban, of the American Uniform Boiler-Law Society (Bulletin No. 7) contains the information that great interest is being manifested throughout the country in the movement looking toward the adoption of uniform boiler laws.

The Industrial Commission of New York State has organized committees for drawing up a boiler code, and Mr. Durban, who has been asked to serve in an advisory capacity says, "We have every hope that New York State will adopt the A. S. M. E. Code." From his personal interviews he has the assurance of the following individuals and officials of their intention of supporting the meeting and general movement.

The Louisiana branch of the engineering society passed a resolution asking the governor to appoint an official representative to the Ohio meeting, and in view of recent legislation passed by that state, it is fair to assume that Louisiana will have a representative there.

Governor Ferguson of Texas has agreed to appoint a representative to attend the Ohio meeting.

The City of Little Rock, Ark., is considering the adoption of a boiler code and will be represented at the meeting in addition to the appointee by the governor for the state.

Commissioner McKelvey, of St. Louis, said that the city would be represented at Washington. The State of Illinois and the City of Chicago will both send delegates.

Detroit will probably send a representative.

Pennsylvania and several of its largest cities will send delegates. Because of the peculiarity of the legislative enactment, cities of the first, second and third class of that state have power to make their own regulations regarding boilers.

The Ohio commission is receiving many acceptances to the meeting where speakers of national reputation will take part, urging the advisability of all states adopting the same standard in the interest of uniformity and urging legislation as a safety measure.

In Tennessee there is a determined effort to have the Code adopted by the state. A full-page illustration has been reproduced from "Power" to show the disastrous effect of an explosion that occurred in Jackson, Tenn., recently and to convince legislators of their duty toward their citizens in passing safety boiler legislation.

ENGINEERING AFFAIRS

The Combined Engineers' Associations of Brooklyn, N. Y., will hold their annual entertainment and ball on Saturday, Nov. 18, at Arcadia Hall, Halsey St. near Broadway, Brooklyn.

The Buffalo Section of the American Society of Mechanical Engineers will hold a meeting on Nov. 29, at which C. H. Bierbaum will give a talk on his recent research work on graphite.

The Irish-American Engineers' Association held its annual entertainment and reception at Prospect Hall, Brooklyn, N. Y., on Saturday, Nov. 4. It proved to be the biggest event in the history of the association. A pleasing entertainment preceded the dancing.

Society of Automobile Engineers—The Metropolitan Section will hold a meeting on Thursday evening, Nov. 16, at the Automobile Club of America, 54th St. near Broadway, at which C. F. Kettering, of Dayton, Ohio, will speak on "Some Present and Future Scientific Problems of the Automobile."

The Combined Manhattan, Bronx and Queens Associations of the N. A. S. E. held their fifteenth annual entertainment and reception on Saturday, Nov. 4, at Terrace Garden, New York City. All the leading national officers and many past national presidents were present. There was the usual large attendance. The entertainment was greatly appreciated, and a program of dancing concluded a pleasant night. The combination comprises the following: Brotherhood, No. 1; James Watts, No. 7; Progressive, No. 15; Protective, No. 23; Phoenix, No. 24; Manhattan, No. 25; German-American, No. 29; Empire, No. 32; Stevenson, No. 44; Mott Haven, No. 47; Insurance, No. 56.

BUSINESS ITEMS

The Clarage Fan Co., of Kalamazoo, Mich., has opened a branch office in charge of L. O. Monroe at 123 West Madison St., Chicago, Ill.

The Rainville Co., of Grand Rapids, Mich., makers of leather belting announce the opening of a New York branch at 6 Church St. under the direction of George S. Baker.

The Nordberg Manufacturing Co., of Milwaukee, Wis., announces the appointment of H. W. Dow as sales manager. Mr. Dow has been associated with the Nordberg Manufacturing Co. in the engineering and sales departments for 12 years.

The Jeffrey Manufacturing Co., of Columbus, Ohio, has opened an agency with the Gustin-Bacon Manufacturing Co., of Kansas City, in order to establish closer coöperation with their customers in Kansas, Missouri and Oklahoma, and improve their service there.

A line of Jeffrey transmission machinery, chains, buckets, etc., will be carried in stock.

"Power Notes," the house paper of the Diamond Power Specialty Co., Detroit, manufacturer of Diamond soot blowers, has just been increased in size. The object of this little four-page monthly paper is to bring to the attention of the consulting engineer, steam engineer, mechanical engineer and architect the best information available on the phases of power-plant work, with which this company is particularly concerned. Records of tests and photographs of installations are run. The Diamond Power Specialty Co. will take pleasure in placing the names of engineers interested in power-plant development upon the mailing list, and in forwarding to anyone who requests it a loose-leaf folder in which the monthly copies may be filed for reference.

TRADE CATALOGS

Ice Harvesting Equipment. Gifford-Wood Co., Hudson, N. Y. Bulletin No. 24. Pp. 40; 6x9 in.; illustrated.

Ball Bearing Hangers. Gurney Ball Bearing Co., Jamestown, N. Y. Catalog H-2. Pp. 40; 8½x11 in.; illustrated.

Class M Volute Centrifugal Pumps. Erie Pump and Equipment Co., Erie, Penn. Catalog No. 32; pp. 32; 6x9 in.; illustrated.

Toledo Pipe Threading Devices. The Toledo Pipe Threading Machine Co., Toledo, Ohio. Catalog. Pp. 16; 3½x6 in.; illustrated.

Diamond Soot Blowers for Water-Tube Boilers. Diamond Power Specialty Co., Detroit, Mich. Bulletin No. 119; pp. 16; 8x11 in.; illustrated.

Vertical Self-Oiling Engines. Engberg's Electric and Mechanical Works, St. Joseph, Mich. Bulletin No. 300; pp. 16; 6x9 in.; illustrated.

Jenkins' Automatic Graphite Grease Lubricator for Valves and Cylinders. The Jenkins Graphite Lubricating Co., Third and Washington St., Louisville, Ky., Catalog. Pp. 18; 6x9 in.; illustrated.

Chloride Accumulator Storage Batteries for use in 32-, 110- and 120-volt isolated lighting and power plants. The Electric Storage Battery Co., Philadelphia, Penn. Handbook CI; pp. 14; 6x9 in.; illustrated.

Water-Tube Boilers. E. Keeler Co., 50 Church St., New York. Catalog; pp. 46; 7½x10½ in. This shows in detail construction of water-tube, cross-drum and horizontal-return tubular boilers built by this company.

Pocket CO_2 Indicator—American Type. Bacharach Industrial Instrument Co., Pittsburgh, Penn. Bulletin No. 4; pp. 4; 6x9 in.; illustrated. Illustrates an improved and simple device for determining the amount of CO_2 in the flue gases.

Lagonda Boiler Room Specialties. The Lagonda Manufacturing Co., Springfield, Ohio. Booklet T2. Size, 3½x6 in.; illustrated. This shows the different types of boiler-tube cleaners and other specialties made by this company.

Handling Coal and Ashes at the William H. Grundy Plant and "Handling Coal in the Victor Power Plant" are the titles of two booklets issued by the Link-Belt Co., Chicago, Ill., showing the use of Link-Belt equipment in these plants.

Ammonia Fittings and Supplies. York Manufacturing Co., York, Penn. Catalog. Pp. 158, 6x9 in.; illustrated. This catalog is exceptionally well illustrated and contains tables and special information which ought to be of value to those interested in refrigeration.

Pipe Fittings, Etc. Pittsburgh Valve and Fittings Co., Barberton, Ohio. Catalog D. Pp., 356; 5x7½ in.; illustrated. This describes a complete line of iron pipe fittings, brass and iron valves, cocks, etc., for steam, gas, water and oil; tabular data are included and prices listed.

Hot, Soft Water for Steam Boilers. Harrison Safety Boiler Works, Philadelphia, Penn. Publication No. 68. Pp. 56; 6x9 in.; illustrated. This describes the Sorge-Cochrane hot-process water softener, which takes advantage of the fact that chemical reactions are more rapid and complete in hot water than in cold.

Encyclopedia of Goodyear Mechanical Rubber Goods. The Goodyear Tire and Rubber Co., Akron, Ohio. This is the title of a set of bound and well-illustrated bulletins containing tabulations and information as to use of the mechanical rubber goods made by this company; these include belts, hose, packing, pump valves, etc.

Proposed Construction

Ark., Dumas—Improvement District No. 2 receiving bids Nov. 28 for electric-light plant. About $25,000. H. W. Wright, engineer.

Fla., Okeechobee—Southern Utilities Co., Jacksonville, has franchise to install electric-light and power system.

Idaho, Coeur d'Alene—Central Heating and Lighting Co. plans expenditure of $100,000 for heating and power plant.

Iowa, Centerville—Centerville Light and Traction Co. plans transmission line from Centerville to Allerton.

Ill., Mt. Sterling—City votes Nov. 15 on bonds for electric-light plant.

Ind., Anderson—Bd. Pub. Wks. plans to improve electric-light plant. About $50,000.

Iowa, What Cheer—Burlington Railway and Light Co. granted 25-year franchise to supply city with electricity.

Kan., Buffalo—City voted $15,000 electric-light plant bonds.

Ky., Uniontown—Kentucky Utilities Co. purchased lighting plant, and will build transmission line from Uniontown to Morganfield. Harry Reid, Lexington, president.

Md., St. Michaels—City plans to install electric-lighting system.

Minn., Ely—Kawishiwi River Power Co. organized to build hydro-electric power plant. About $1,500,000. T. W. Orbison, Appleton, Wis., engineer.

Mo., Brunswick—Brunswick Light and Water Co. plans to improve lighting system. L. A. Nickel, manager.

Mont., Poplar—City voted $6,500 bonds for electric-lighting plant.

N. Y., Caledonia—Nedrew Manufacturing Co. have made plans for boiler house.

N. C., Bethel—City to rebuild electric-light plant.

Okla., Pawhuska—City to vote on $7,000 electric-light bonds. J. M. Buckley, city engineer.

Ohio, Cleveland—G. C. Kuhlman Car Co. granted permit to improve boiler house, 760 East 140th St. About $10,000.

Ohio, Cleveland—National Artificial Silk Co. granted permit for power house, 9801 Wolford Rd. About $55,000.

Panama—Major E. I. Brown, general purchasing officer, Panama Canal, Washington, D. C., will receive bids, Nov. 24, for steam cocks, gate valves, brass check valves, brass globe valves, hydraulic gage, rubber valves, gaskets for miter gates, copper cable, wire, bell ringing transformers, watt-hour meter, electric bells, galvanized outlet boxes and copper cable connectors.

Penn., Philadelphia—A. L. Fretz & Son plan power house at plant on Wilde St.

S. D., Bruce—City plans election to vote on $7,000 bonds for electric-lighting system.

Tenn., Nashville—St. Thomas Hospital plans brick power house. About $22,000.

Tex., Bertram—Elias Allen plans electric-light and power plant.

Wis., Gresham—Gresham Electric Light and Power Co. granted franchise for electric-light and power plant. L. D. Laudenstein, village clerk.

Wis., Balsom Lake—Dahlberg & Son will install electric-light system for city.

Wis., McDill—J. Strange Paper Co., Neenah, plans dam and hydro-electric plant at McDill, near Stevens Point.

Wis., Stevens Point—Wisconsin Valley Electric Co., Wausau, purchased local electric and gas plant and will extend power lines to city